COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Vol. 10, No. 2 AUGUST, 1938 25c a copy

Construction view of furnace in one of two high-pressure units now nearing completion

Steam and Feedwater Controls at Springdale

Factors Involved in Selection of Mechanical Draft Fans

High-Pressure Steam Generators

COMBUSTION
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VOLUME TEN NUMBER TWO

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FOR AUGUST 1938

FEATURE ARTICLES

EDITORIALS

DEPARTMENTS

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Combustion is published monthly by Combustion Publishing Company, Inc., a subsidiary of Combustion Engineering Company, Inc., 200 Madison Avenue, New York. Frederic A. Schaff, President; Charles McDonough, Vice-President; H. H. Berry, Treasurer; G. W. Grove, Secretary. It is sent gratis to consulting and designing engineers and those in charge of steam plants from 500 rated boiler horsepower up. To others the subscription rate, including postage, is $2 in the United States, $2.50 in Canada and Great Britain and $3 in other countries. Single copies: 25 cents. Printed in U. S. A. Publication office, 200 Madison Avenue, New York. Issued the middle of the month of publication.

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EDITORIAL

Not So Important

Additional PWA funds, aggregating over twenty-eight million dollars, have lately been released by Secretary Ickes to cover the cost of forty-five municipal power projects. These grants have received wide publicity in the daily press. Examination of the list shows that about half are non-competitive with existing facilities and the remainder fall under the now famous formula of becoming competitive "only after reasonable efforts, made in good faith, to acquire the existing local utility, have failed."

This amount although considerable, in itself, is small compared with the untold millions that are being expended on federal hydro projects, the rural electrification program and other special power allotments.

While these latest manifestations of power consciousness and leanings toward public ownership on the part of a paternalistic government will be deplored by some, it is well to remember that they apply mostly to small communities and involve diesel or steam units of relatively small capacities; in some cases they represent extensions to existing plants or distribution systems. As such they are of less importance than might appear at first.

A Definition of pH

Engineers are accustomed to employ certain rules and symbols in the routine solution of their problems. Although fully conversant with the use of such terms and possessing a general knowledge of their significance, it is believed that many would falter if called upon to define them. One such term which is in daily use in power plant work is pH value. We all know, of course, that this relates to hydrogen ion concentration, and that pH = 7 represents the neutral point, values above or below which designate alkaline or acid conditions, respectively. But just why is this so?

This appears to be answered very clearly in an unidentified clipping from some dentists' circular or paper which was recently handed to the editor. It has the following to say about pH:

"As is most generally known, the symbol pH with a number written after it, represents the concentration of the hydrogen ion in a given solution. But what is not generally known, is the fact that this same symbol is simply an abbreviation of the term '*inverse* logarithm of the hydrogen ion concentration.' In other words, the lower the pH of a solution, the greater is its acidity.

"For example—by actual experiment, we find that a given solution dissociates into ions to the degree of one part in ten thousand. We therefore express the dissociation as (H) 1/10,000 or $1/10^4$, and by taking advantage of a simple rule of exponents 10^{-4}. Now the logarithm of the last number is (-4), but to simplify it, we use the inverse logarithm (the symbol of which is 'p') and we may now express the hydrogen ion concentration as p(H) 4.

"The reverse is true of alkaline solutions. They possess a lesser relative amount of hydrogen ions and therefore the inverse logarithm representing the dissociation, will be greater. For example—the dissociation of a given solution is found to be one part in one hundred million. In other words, (H) is equal to 1/100,000,000 or $1/10^8$ or 10^{-8} or p(H) 8. The dissociation of pure water is one part in ten million and so its pH is 7. This is considered the point of neutrality."

Enlightening the Public

The president of the American Society of Heating and Ventilating Engineers has named an industry-wide committee on public relations to clarify the public conception of air conditioning and define the service which users can reasonably expect from various types of systems now in use. The program is educational and is designed not only to enlighten present and prospective users of air conditioning, but also to combat misleading claims by unscrupulous elements which have thrived on the growth of air conditioning during recent years.

This is a most commendable idea. Would that some similar organization undertook to enlighten the public as to the truth about water power.

Performance vs. Normal Operation

It is generally conceded that the best performance is obtained with a new automobile after it has been run a thousand or fifteen hundred miles and then tuned up.

If this be true of a highly standardized and exhaustively tested product such as an automobile, what of the power plant which contains many pieces and kinds of equipment, few if any standardized, and each of which must function both individually and in coordination with other equipment under variable conditions. It is not surprising, therefore, that considerable time is often required to rectify initial troubles, make adjustments and permit the operating force to become familiar with the characteristics of the new equipment. Also, it is important that the normal operating conditions, upon which designs were predicated, be established before performance is judged. For instance, it is not unusual for a new steam generating unit to show low superheat during initial operation. This may be due to the clean surfaces ahead of the superheater absorbing more heat than would be the case after a certain equilibrium coating of slag has built up on the furnace walls during normal operation.

Other similar instances could be cited to emphasize that it is always well to defer tests until normal conditions have been reached not only with the unit under test but also related equipment that may have bearing on the results.

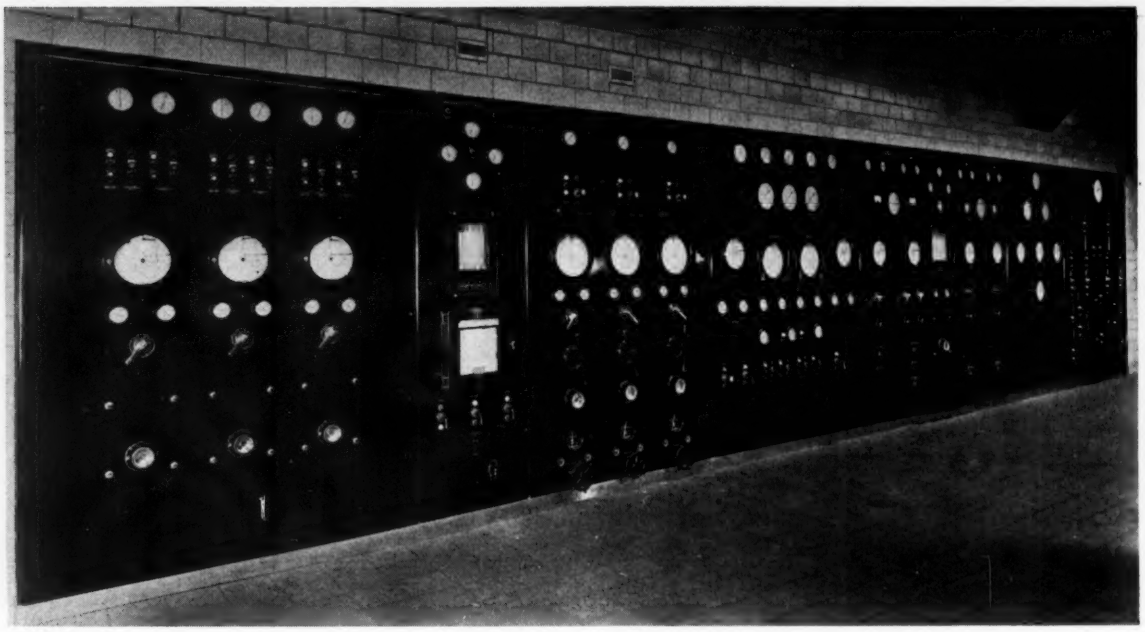

Fig. 1—Group of control panels
Viewed from left to right the numbering of these panels as referred to in the text is 6, 4, 5, 1, 2, 3 and 7

Steam and Feedwater Controls at Springdale

The three high-pressure boilers which supply a 50,000-kw topping turbine at the Springdale Station of the West Penn Power Company, near Pittsburgh, are provided, in addition to the water-level control, with an excess pressure pump control, a pump bypass control to prevent overheating at low loads, and pressure-reducing and desuperheating equipment for operation at such times as the high-pressure turbine is off the line. All these controls are automatic although manual control is also provided. The combustion control and high-pressure turbine-exhaust desuperheater control are also briefly described.

At the Springdale Station of the West Penn Power Company boiler water for the three new 1450-lb high-pressure steam generating units is supplied by four centrifugal pumps, each of 1554 gpm capacity at 3600 ft head. Two of these pumps are driven by steam turbines, one by a motor and the fourth has a dual motor and turbine drive.

These turbines take steam at 340 lb abs and exhaust at 100 lb abs to two heaters which further heat the feedwater to 330 F before it enters the high-pressure boilers. It is necessary to supply additional live steam for the heating cycle at all times and especially when the turbine loads are light. This is accomplished by two reducing valves set to cut in as the heater pressure drops to a predetermined point and introduce live steam at reduced pressure into the heaters.

Boiler Feed Pump Control

In order that a constant excess of feed pressure over steam pressure may be maintained at all boiler ratings, and to permit various combinations of the four feed pumps to operate automatically in parallel, an excess-pressure control system has been installed. The group is under the control of a master controller which communicates loading pressures to standard Smoot series regulators on the pumps, in accordance with variations in the excess of boiler feed pressure over steam pressure. On each turbine-driven pump the regulator controls the pump speed by actuating valves in the turbine governor oil circuit, whereas on the motor-driven pump the regulator actuates a valve in the discharge line. On the dual-driven pump both of these two methods are employed, according to whether the pump is being driven by the turbine or the motor.

The control system is based on a differential pressure which, through the master controller and the regulators, measures the flow through each pump and adjusts the output to give the correct total flow of water and maintain the desired excess pressure. This is accomplished by placing a flow nozzle in the suction line of each pump, the differential drops across these nozzles being communicated to the diaphragms of the pump regulators. This differential, which is a direct measure of the flow through the pump, is balanced on the weigh beam against the loading from the master controller. The latter, in turn,

reflects the differential pressure across the steam and feedwater headers. Ratio adjusters on the master controller permit the master loadings to be varied in accordance with the desired distribution of load between the pumps. Means are provided for transferring the pumps from automatic to manual control, if desired, by a transfer lever and an adjusting wheel. The control arrangement is represented diagrammatically in Fig. 2.

On each of the boilers, water level is controlled by a standard Smoot water-level control system.

Pump Bypass Control

Another important feature of the regulation of these boiler feed pumps is a pump bypass control which is designed to prevent overheating and consequent possible damage to the impellers and bearings at low loads. This bypass control removes some of the water from the pump discharge and flashes it through an orifice to the deaerators during periods when the flow reaches a predetermined low value and beyond which there is danger of overheating.

A bypass valve is installed on the discharge line of each pump under the control of the flow meter, located in the suction line, and when the point of low flow is reached the meter operates a contactor to control a solenoid valve which opens the bypass valve. The amount of recirculation is determined by a throttling orifice in the bypass line downstream from the bypass valve. However, when the flow through the pump begins to rise above the point at which the bypass opened, the latter does not close immediately. Instead, the control delays action until the flow has increased to a second predetermined value, whereupon a high-flow contactor is actuated by the flow meter to operate the solenoid valve and close the bypass. This action produces an overlap and prevents hunting by eliminating a critical on-and-off control point. A recorder shows on a single chart the total flow of water through the pump, the time at which the bypass opens, the duration of its opening and the time it closes.

The bypass valve, itself, is of the single-seated turbine-type, cylinder-operated. It is held closed by air pressure and opened by a strong spring. The solenoid valve admits air pressure to the top of the operating piston to hold the valve closed against the spring. When the flow meter actuates the solenoid valve to open the bypass, the solenoid valve closes, shutting off the air pressure and allowing the spring to open the bypass. Should the air pressure fail or the electrical circuit be interrupted the bypass will be opened by the spring. This provides a safety measure.

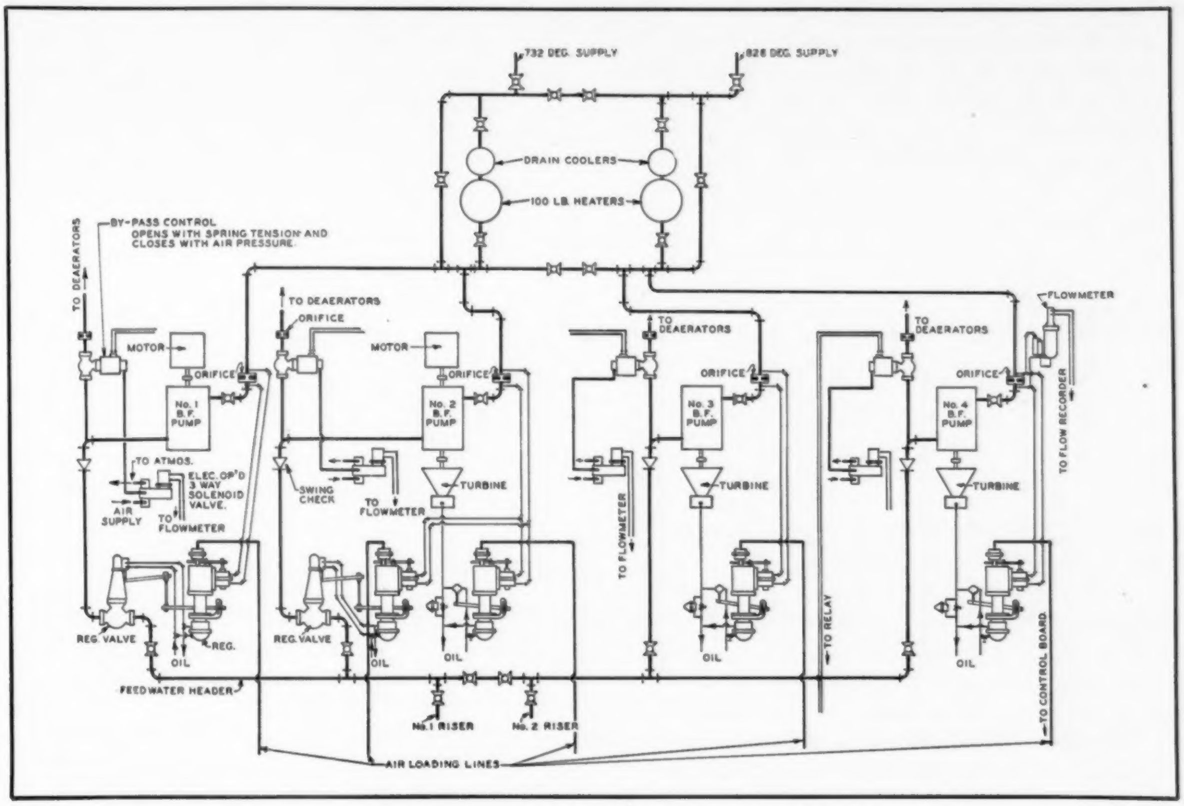

Fig. 2—Diagram of boiler feed pump control

The flow meter which actuates this control is a standard Republic electrical type with high- and low-flow contactors and such additional relays and electrical circuits as are necessary to operate the solenoid valve. It has a two-pen recorder, one pen recording the total flow through the pump and the other the full opening of the bypass valve.

Each of the three high-pressure boilers at Springdale is rated at 525,000 lb per hr maximum output with 330 F feedwater and steam is supplied at 1250 lb, 935 F to a 50,000-kw turbine which exhausts at 340 lb to the low-pressure units.

Superheat is controlled by dampers at the boiler outlet, which permit bypassing some of the gas around the superheater. The operation of these dampers is controlled by a Bailey temperature controller which receives a pressure response due to a temperature change at the superheater outlet, this response being transmitted to the regulator which operates the dampers.

Desuperheating Exhaust at Reduced Load

The high-pressure turbine is provided with an Elliott cartridge-type desuperheater for use in reducing the steam temperature of the exhaust under conditions of reduced turbine capacity. As it is only necessary to desuperheat the exhaust during these periods of low load it is obligatory to control the operation of the desuperheater. Bailey meter control for the exhaust desuperheater is designed to function with either a drop in load or a temperature rise. Either or both may cause the operation but normally the system functions with a sudden drop in load equivalent to approximately one-half capacity. When this occurs an arm on the load controller engages a pin on the turbine governor rod and a setting is made in anticipation of the bulk of the desuperheating necessary. A temperature controller finally administers the micrometer adjustment and continues to adjust the temperature for small changes.

The control apparatus consists of two temperature controllers, one for each of the two branch lines in the turbine exhaust. The higher temperature in either case takes over the control. In addition, there are the necessary averaging relays, pressure switches and water valves. It is possible with a bypass connection to operate the water supply to the desuperheater cartridge manually.

Pressure Reducing and Desuperheating System

Should the high-pressure turbine be out of service, the steam from the high-pressure boilers must be reduced in pressure and desuperheated in order to supply the low-pressure turbines direct. The Smoot pressure-reducing and desuperheating equipment that has been installed for this purpose is shown in Fig. 3.

It will be noted that there are three bypass lines in parallel, each capable of handling the maximum output of its boiler. These are controlled automatically by a master controller so that each can pass the same or different proportions of the total load. The pressure-reducing valve in each line, which is of the turbine type, has an 8-in. inlet and a 12-in. outlet, its opening being controlled by an individual regulator that receives its loading pressure from the pressure-reducing master controller. These valves are designed so that the required pressure reduction will be from 1275 lb to 340 lb.

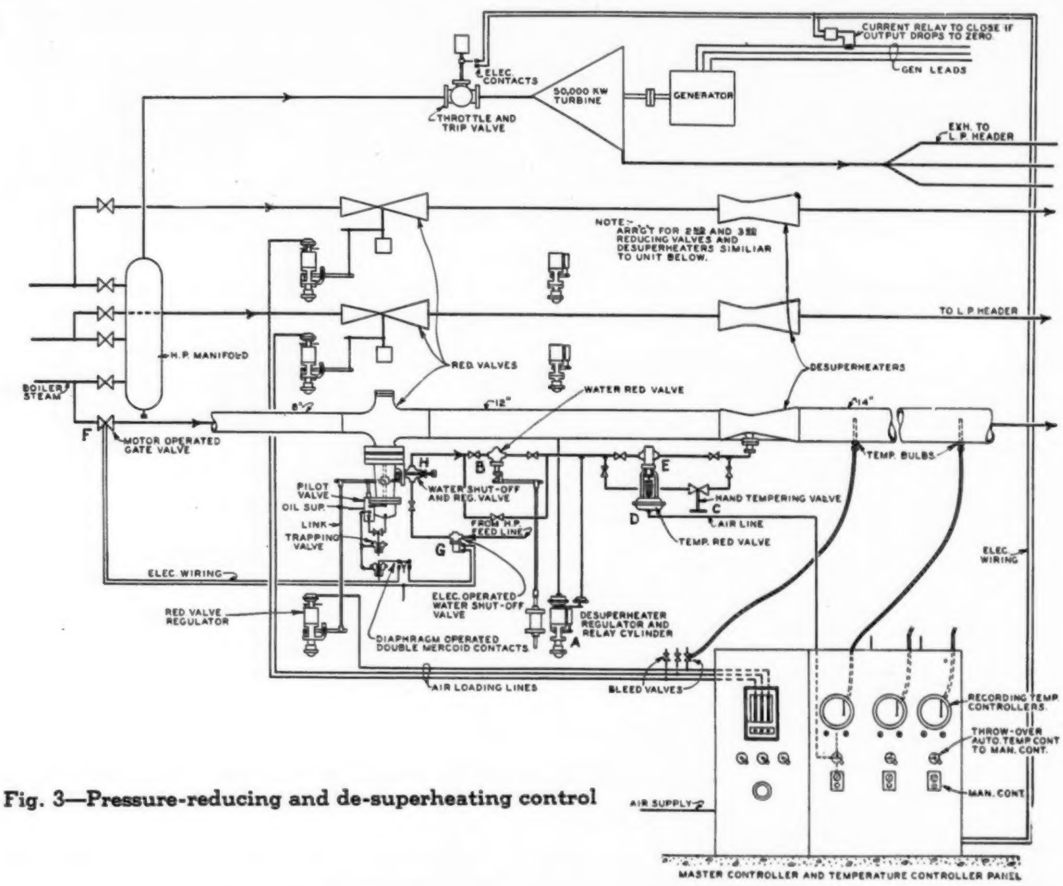

Fig. 3—Pressure-reducing and de-superheating control

The desuperheater in each of the three lines is of the spray type with ratio and temperature control, the amount of water supplied to the spray nozzle being controlled by regulator *A* which is governed by the pressure drop across the venturi throat of the desuperheater. This regulator operates a water-ratio valve *B* in the water line so that the flow will be in direct proportion to the flow of steam through the desuperheater. After passing through this water-ratio valve, the water flows through the hand-tempering valve *C* which is set to produce the required temperature reduction from 925 F to the desired temperature on the down-stream side of the desuperheater.

In order to compensate for variations in the desuperheating water temperature, as well as for variations in the high-pressure steam temperature a temperature controller *D* is placed in parallel with the hand-tempering

valve C and actuates the valve E, thus readjusting the flow water for the exact low-pressure steam temperature desired. This temperature controller is actuated by air pressure from the master control panel.

Several important refinements have been applied to the control of these parallel bypasses. On each one, a motor-operated steam valve, F, in the steam header ahead of the pressure-reducing valve is electrically connected to another motor-operated water shut-off valve, G, in the desuperheater spray water line so that the two open and close in synchronism. A control device connected in the oil power circuit of the pressure-reducing valve operates to close the motor-operated valves and shut off both steam and desuperheating water completely in the event of failure of the oil pressure. There is also provided a trapping valve that will maintain the reducing valve in the position it occupied when the oil pressure failed.

A second water shut-off valve, H, built integrally with the pressure-reducing valve, is mechanically operated by a link from the reducing valve stem and will close if the reducing valve closes.

As another safety precaution, a temperature safety controller is connected to operate a leak-off valve in the master loading line leading to each reducing valve regulator. This controller is set so that, if the low temperature rises to a predetermined high value, it will open the leak-off valve and change the master loading on the reducing valve regulator so that the valve will close to reduce the steam flow through the bypass until the low temperature has returned to normal. The temperature controller will then close the leak-off valve and the regulator will operate under the normal loading pressure transmitted from the master controller.

Valve and Desuperheater Materials

The reducing valves and desuperheaters are built of carbon-molybdenum steel, the valves being built to the 1500 lb A.S.A. standard and the desuperheaters to the 600 lb A.S.A. standard. Carbon-molybdenum steel is used in the desuperheater because the high temperature exists almost to the downstream side of the desuperheater even though the pressure at this point is lower.

On both the pressure reducing and desuperheating panels, standard transfer valves and levers are provided for transferring the bypass equipment to manual control. While the bypasses are under manual control, the topping turbine might trip out or the high- and low-pressures might vary so much that the high-pressure safety valves would pop or operation of both high- and low-pressure turbines be affected. Accordingly, on the back of the controller panels of the pressure-reducing and desuperheating system, a number of throw-over devices have been provided that will automatically transfer the controls of both pressure reducing valves and desuperheaters from manual back to automatic, if the high and low pressures vary beyond certain predetermined limits or the turbine trips during manual operation.

These devices consist of a number of steam pressure elements, diaphragms and leak-off valves, balanced on weighbeams in such a way that when the pressure limits are reached, they control the pressure of air in transfer cylinders that contain pistons connected by linkages to the manual-automatic transfer valves. When the air pressure is applied to these transfer cylinders, the movement of the pistons literally takes the control out of the operator's hand and transfers it back to the regulators under control of the master controller.

Control Panels

Operation of this equipment is controlled, indicated and recorded on a group of seven Republic control panels. On the topping turbine control panel (No. 1 see photograph, Fig. 1) steam pressures are indicated by dial-type gages for various points in the turbine and feedwater heating system, while steam flow, throttle pressure and exhaust pressure are shown by round-type recorders. Temperature controllers for throttle and exhaust temperatures are mounted on this panel. Lubricating oil and governing oil pressure, also gland-seal pressures, are indicated and controls are provided for the turning-gear motor and for the motor-operated gate valves. Throttle and exhaust temperatures are recorded and in addition there are two Bailey temperature controllers for the turbine exhaust which maintain the proper temperature at light loads with the cartridge type desuperheater previously mentioned.

On another panel (No. 2) are mounted the master controller of the pump-control system, together with the ratio controls and manual-to-automatic transfer levers. Suction and discharge pressures of each pump are shown by indicating gages, and pump outputs are recorded. There are also ammeters and push-button controls for the pump motors.

On a third panel (No. 3), data on feedwater performance are indicated and recorded, as well as pressures on the 350-lb system, pressures and temperatures at the heaters, etc.

The fourth (No. 4) and fifth (No. 5) panels carry the master controller of the pressure-reducing and desuperheating system together with the manual controls for the desuperheaters and automatic throw-over safety devices, described above, together with the necessary indicating and recording instruments to give information regarding the pressures and steam temperatures at various points in the line. Similarly, controls and instruments for the pressure-reducing valves, including the automatic throw-over safety devices are mounted on panel No. 6.

Finally, to aid the operators in controlling the conditions in the boiler feed piping system, a large diagram of the piping and heater system, using international symbols, is aid out in plated copper strips on a diagrammatic panel (No. 7) with operating signal lamps and push buttons to show the operating condition of each valve and each piece of equipment in the system.

In addition to the foregoing controls for boiler feed pumps, pressure-reducing and desuperheating, brief mention of the combustion control and mill-level control may be in order.

Combustion Control

The Hagan system of combustion control as installed at Springdale is designed to control either manually or automatically conditions on the three high-pressure steam generating units, each of which is fired by three unit pulverizers and each equipped with constant-speed forced-draft and induced-draft fans. The fans have inlet vane control.

The basis of control is the pressure on the steam

header which is set to a certain standard by a master sender. This transmits an air loading pressure to three primary air relays on each boiler and as the primary air is directly proportioned to the coal feed the master sender has control of the pulverized fuel. The forced-draft fan vanes receive their adjustment from the effect of the master sender loading upon the air pressure drop across the air preheater. Furnace pressure is the control medium that is employed for the operation of the induced-draft fan vanes.

A special feature of this combustion control is a set of solenoid-operated air valves incorporated into the induced-draft fan vane-control system so that if one fan becomes inoperative due to mechanical or electrical failure, its vanes are closed and the boiler rating is adjusted to the capacity of the other fan. An additional feature is provision for safety in the event of loss of air pressure in the forced-draft duct or loss of suction in the induced-draft duct below a set minimum. Safety checks in each air duct operate with loss of forced draft to trip the fan motor which, in turn, trips the pulverizers, blowers and feeders, and with loss of induced draft to trip the induced-draft fan motor, which trips the forced-draft fan motor, etc. At the same time if the induced-draft circuit trips, the vanes on each fan are opened wide by the solenoid valves mentioned above and thus the boiler is placed on natural draft and is out of danger.

The combustion control board is 38 ft long and is composed of three boiler panels and one master panel. The master panel contains pressure gages showing master air loading and transfer valve handles to change the control from manual to automatic. In addition, the main steam and feedwater pressures are recorded and the motor control switches for the oil torch are also located at this point.

Each boiler panel contains draft and pressure multi-pointer gages, motor connections and control switches for mills, blowers, feeders and fans. The mill level controllers are also included as well as the automatic superheated safety valve control. The Hagan control is mounted with nothing but air loading pressure gages, transfer handles and manual control knobs showing. Above this control the recording meters are located. These record CO_2, gas, water and superheated steam temperatures, steam air and feedwater flow, furnace pressure and boiler-water level.

Mill-Level Control

The Bailey mill-level control operates on the difference in pressure between the pulverizer differential and the primary air differential. This difference in pressure is set so that changes in it will cause intermittent operation of the coal feeder by making and breaking contacts in the feeder-motor circuit.

Factors Involved in Selection of Mechanical Draft Fans

By W. S. PATTERSON

Combustion Engineering Company, Inc.

A discussion of the numerous influences that must be taken into consideration in calculating the capacities of forced-draft fans, induced-draft fans, overfire-air fans, pulverizer exhausters, vent fans and primary-air fans. Checking lists for such requirements are included and characteristic curves reviewed. The conclusion is that liberal tolerance in fan capacity should be provided in order not to limit the capacity of the steam generating units.

THE problem of specifying the proper operating conditions at various loads on the fans necessary to operate a steam generating unit is one that is generally left to the manufacturer of the steam generating unit. The fan manufacturer assumes limited responsibility for producing the volume and pressure specified to him. The word, "limited," is used because the Fan Test Code requires certain conditions to be maintained on a test which are seldom, if ever, duplicated in an actual installation, thereby making it practically impossible to test a fan under actual service conditions. Furthermore, the connections made to a fan may affect its performance to such an extent that it would be unwise to select a fan without large pressure and volume tolerances unless it be actually tested with inlet and outlet connections, identical to those expected under service conditions.

Many power plant fans are somewhat special to the extent that the wheel may be narrower and of smaller diameter than the standard wheel for the fan size selected from the manufacturer's catalog of standard sizes. Many other fans are entirely special as to both wheel and housing dimensions. The performance of such fans is generally determined by the manufacturer from actual test or from laws as applied to the fan or fans of the series which were tested for actual performance. In purchasing fans which vary from the standard line as outlined above, the purchaser should assure himself that those of the particular widths or proportions have been actually tested by the manufacturer. If this is not done, the purchaser, to assure himself that the fans will operate in agreement with specifications, should specify a witness test at the manufacturer's plant.

The foregoing factors relating to calculation and measurement of fan performance, will indicate to fan users that unless they are willing to bear the expense of an actual test at the manufacturer's plant, it may be difficult to determine whether the fan will operate in agreement with specifications. The factors discussed below which pertain to the calculation of the conditions of operation will indicate further that in any case it is necessary to apply reasonably large tolerances in the selection because of the many variables involved.

Nearly all large units, regardless of fuel and method of firing, employ mechanical means for furnishing the air for combustion and removing the products of combustion, these fans being commonly termed, "forced-draft fan" and "induced-draft fan," respectively. In addition to the regular forced-draft fan, it is common practice when burning bituminous coal on a chain grate stoker to provide for air over the fire by means of a separate "overfire air fan." Pulverized coal firing with the storage system generally introduces three other fans, namely, first: The "pulverizer exhauster" to handle the coal and air mixture between the mill and the collector and to recirculate air back to the pulverizer; second, the "vent fan," generally used to vent air and evaporated moisture from the system; and third, the "primary air fan" which again mixes air with and carries the coal to the burners. The direct-fired system employs the "pulverizer exhauster" to perform all the functions of the three fans used with the storage system.

Forced-Draft Fans

The requirements of a forced-draft fan cannot be accurately and simply calculated from the fuel requirements and the excess air. In the first place, the excess air is generally specified at a point in the system where it can be measured by calculations made from a gas analysis. This point is usually taken as the boiler outlet, and the air weight calculated with excess air specified at this point includes all the infiltration which has taken place through the boiler setting, furnace walls, ashpit, pulverizer, etc. It is apparent, therefore, that an accurate determination of the air quantity actually supplied under control requires a knowledge of the magnitude of the various forms of infiltration. This subject was discussed in detail in previous issues of this magazine.[1]

On an installation having a small tight setting with a minimum of expansion joints, observation openings and access doors, a sealed ashpit or solid floor, and tight ducts and windbox compartments, the air actually supplied under control will more nearly approach the quantity of air as calculated from the excess air at the boiler outlet. But since leakage is hard to estimate and will vary with the life of the setting and with the care exercised by the operating force in keeping doors closed and maintaining minimum allowable draft in the furnace, it is difficult to more than approximate the furnace leakage. However, on installations employing an air preheater, it is necessary to estimate conservatively the infiltration which is not passed through the preheater in order not to over-

[1] See COMBUSTION, February, April and May 1934.

guarantee heater performance; but this conservatism with respect to the heater means the reverse with respect to estimated fan capacity.

The same problem arises when we attempt to allow for the tempering air used in a pulverizer for direct firing. The air heater performance will generally be estimated on the basis of using fuel of normally low moisture, and, with high preheat, a large quantity of tempering air may be required. This will generally be drawn direct from the room by pulverizer suction and therefore will not pass through either heater or fan. But the equipment will be required to operate satisfactorily with perhaps much higher moisture coal at certain times of the year, depending on coal storage facilities. Under such conditions, less than normal tempering is required and the fan must be selected to meet these extreme conditions.

Separate Fans for Overfire Air

As mentioned, air is usually supplied over the fire with bituminous coal when burned on chain grate stokers. In order to have ample pressure available at the overfire-air nozzles at low loads it is desirable to use a separate fan. When no preheater is used, this fan may draw its air from the boiler room, thus reducing the required capacity of the regular forced-draft fan. When a preheater is used, the overfire-air fan may be used as a booster fan, taking air from the preheated air duct or direct from the room. In any case, it is questionable whether the overfire air should be subtracted from the total in selecting the forced-draft fan, even though it may reduce the required capacity under certain conditions of operation, because it may be found advantageous at the highest loads to operate without overfire air. This depends to some extent on the operator and to a large extent on the design of the furnace arches and location of the overfire-air nozzles.

When direct firing of pulverized fuel is used without an air heater, all the primary air will usually be drawn from the room and may be subtracted from the total for determination of forced-draft fan requirements.

On storage system units where preheated air is used from the main preheated air duct for drying in the pulverizer but is not returned to the system by venting to the primary air fan or furnace, the quantity of air so used must be added to the combustion air handled by the forced-draft fan.

The possibility of outward leakage from the ducts and compartments under pressure must not be ignored. Duct joints, expansion joints, damper rod openings, air heaters, wind boxes of burners and stokers are all possible sources of outward leakage unless carefully designed, fabricated and erected with this in mind.

The use of air-cooled walls surrounding the furnace introduces other considerations, depending on whether the walls are under suction or pressure. If the forced-draft fans discharge air through air-cooled walls, there may be considerable loss of air due to direct leakage into the furnace and boiler room, but at the same time a lesser quantity of infiltration air passing from the room into the furnace. Therefore, the air to be handled by the fan will be increased by the amount of direct leakage to the boiler room and by the amount of the reduction in furnace infiltration. On the other hand, when the forced-draft fan is exhausting air from an air-cooled wall, there will be infiltration from room to air stream, from furnace to air stream and a lesser amount than normal infiltration from room to furnace; all three of these factors tend to increase the quantity to be handled by the fan.

The forced-draft fan will usually handle room air. There may be considerable variation in its temperature depending on the season and on the location of the fan. In general, stoker-fired jobs without air heaters are most likely to have fans in a cool basement or operating floor location whereas a location near the roof, often necessitated by the location of fan in close proximity to the air heater inlet, may result in ambient temperatures as high as 150 F. The temperature and elevation of the fan above sea level both affect the density of the air, which in turn affects the capacity of the fan. These factors must be taken into consideration if the fan requirements are to be predicted accurately and the correct temperature and barometric pressure should be specified and used in calculating the volume to be handled.

The volume calculated in accordance with the foregoing gives the actual required fan capacity under good conditions of operation, with assumed excess air, steady load, tight ducts, normal fuel, etc., such as might be expected during an acceptance test. This volume together with the correct pressure necessary to force the air through all the resistances imposed is useful for evaluating power consumption at the various loads, selecting control equipment and in selecting a fan to give its maximum efficiency at the desired normal rate of output of the steam generating unit, but these actual calculated operating conditions should not be used as maximum requirements in purchasing the fan. The outward leakage corrections may be exceeded or the infiltration actually less than estimated; or it may be necessary and desirable at times to operate with more excess air; or the actual temperature at the fan may be higher than anticipated; or lastly, the fan may not be up to expectations because of poor inlet and discharge conditions. It is therefore well to specify 25 per cent excess volume in selecting a forced-draft fan but this figure may, of course, be modified if extreme conservation was used in arriving at the so-called "actual" requirements, or if they were calculated for an unusual peak load well above the desired maximum continuous rate of operation.

Since forced-draft fans usually have open inlets, the required static pressure is the sum of all series resistances between the fan outlet and the furnace plus or minus allowance for change in elevation between preheater and burner or stoker windbox when preheated air is used. The required windbox pressure at the burner or stoker cannot always be accurately predicted. If the boiler unit is to operate for long periods at a steady load, close adjustments can be made and the required pressure may be taken as that determined from acceptance tests of similar apparatus under identical conditions of operation. But to allow for the variation due to good and poor operators, fluctuating loads, coal sizing, coking and caking tendencies, etc., it is necessary to specify higher pressures for purposes of fan selection, and since the stoker or burner manufacturer does not always furnish or have control of the selection of the fan, it is best for him to set up an excess pressure allowance in the individual item of windbox pressure.

It is also difficult to predict accurately what the windbox pressure variation will be with variation in load since

the fuel bed of a stoker and the air registers of the burners are not fixed resistances. Variations in fuel bed thickness, length and uniformity are all factors which make it difficult to predict the windbox pressure of a stoker. And with pulverized coal burners, the vanes or dampers are adjustable or burners may be cut out as load decreases. Therefore, the system resistance of a forced-draft fan may not vary as the square of the quantity of air being handled and, especially with pulverized coal, gas or oil firing, may not be a smooth curve over the range of loads desired.

Since 25 per cent excess volume was recommended for the forced-draft fan, at least 50 per cent excess pressure should be applied to all fixed resistances in the circuit such as cold air duct, air preheater and hot air duct in the selection of the fan; or if the excess volume is reduced for reasons mentioned above, the excess pressure may be reduced accordingly provided, of course, the actual pressure requirements were calculated on the basis of passing the more conservative actual air requirements through the system.

TABLE I—CHECK LIST OF ITEMS FOR CALCULATION OF FORCED-DRAFT FAN REQUIREMENTS

		Volume
1.	Excess Air	Is it conservatively specified or absolute minimum?
2.	Infiltration	Boiler setting, furnace walls, doors and joints, ashpit, pulverizer.
3.	Leakage	From ducts, windboxes, air-cooled walls, air to gas in air heater.
4.	Tempering	Deduct only the tempering needed for maximum moisture in coal.
5.	Furnace Draft	Its magnitude affects furnace infiltration.
6.	Over-fire Air	Consider source (room or air duct) and necessity at maximum load.
7.	Pulverizer Air	Is all, part or none first handled by forced draft fan? In storage system, consider also ultimate disposition.
8.	Air-cooled Walls	Affect of pressure or suction on leakage and infiltration.
		Temperature and Density
9.	Position	Will fan be in basement, under roof, near windows, etc.?
10.	Air Source	Will fan draw air through ducts from hot location or hollow walls?
11.	Elevation	Lower density air increases vol. and system resistance; and decreases fan performance.
12.	Season	Seasonal fluctuations particularly important with outdoor locations.
		Pressure
13.	Inlet	Will fan have inlet boxes and what negative draft?
14.	Discharge	Add all series resistances; are they conservative or minimum?
15.	Evasé	Properly proportioned outlet will recover velocity pressure.
16.	Windbox	Has stoker and burner manufacturer already added excess allowance.
17.	Elevation	With preheated air, allow for "stack effect."
18.	Burners	Allow for shutting off burners at reduced load.

Induced-Draft Fans

The induced-draft fan generally follows all the heat recovery apparatus and may follow also apparatus for dust elimination. In some instances, it is furnished as an integral part of the stack, built right into the base. This fan must, therefore, handle the gases resulting from the combustion of the fuel and also all the air infiltration occurring up to the fan inlet, including direct air leakage in the air heater. It may also handle cinders and fly ash. The infiltration may amount to as much as 20 per cent of the total weight of gases being handled since it is not uncommon to allow 1 per cent reduction in CO_2 across each large piece of apparatus operating with a high differential pressure across the enclosing walls.

The volume requirements of induced-draft fans are figured at the calculated temperature and density existing at the fan inlet. The density should be corrected for elevation above sea level and for the moisture in the flue gas which is considerable with some fuels, such as wood and natural gas. An allowance must be made in either the density or the excess capacity factors to allow for the fly ash.

The weight of gas from which the volume at the fan is calculated is generally determined by assuming from experience the expected increase in excess air across each piece of apparatus and connecting ductwork between furnace and fan inlet, and using the calculated fuel requirements. This gas weight must include the moisture from fuel, from combustion of hydrogen, and from the air or any other source such as water seals or jets in ashpit.

The dry flue gases produced from the burning of fuels under the usual combustion conditions have a combined specific volume which is less than that of air at the same temperature. The moisture existing at a partial pressure as superheated steam has a specific volume about 1.6 times as great as dry air. From this ratio or by the use of steam tables, the true specific volume of the gas and moisture mixture may be determined, but for ordinary fan calculations the error will be of low magnitude if the specific volume of wet gas be taken as equal to the specific volume of dry air at the same temperature. The volume per pound thus determined multiplied by the weight of wet gas estimated at proper fan inlet excess air will give the actual capacity requirement of the fan.

The fly ash carried in the gases from a pulverized-coal-fired unit may be as high as 80 per cent of the ash fired with the coal unless efficient dust recovery apparatus is placed ahead of the fan, but this ash does not appreciably increase the total volume of the mixture handled because the density is increased by the ash. The effect on the fan performance of this increased density is to produce a higher static pressure than would be expected with dust-free gas, assuming the fan efficiency is not decreased by an accumulation of ash. The presence of the ash may, therefore, increase the power requirements of the drive for a given speed and capacity. However, since the ash carried will generally be less than five grains per cubic foot, no allowance need be made other than the excess factors recommended below.

As in the case of the forced-draft fan, excess capacity should also be allowed to take care of operation with higher excess air than is possible under test conditions, and lowered fan performance due to poor inlet conditions and worn or dirty fan blades. To allow for these factors, it is, therefore, well to specify at least 20 per cent excess volume in selecting the fan unless the actual operation requirements were calculated very conservatively or were figured for a peak load of short duration.

Induced-draft fans usually operate against a negative static pressure at both inlet and outlet although the pressure at the outlet may be balanced or sometimes positive when there is a long breeching connection to a short or overloaded stack, or when the control damper is at the outlet. Positive pressure at the fan outlet is generally undesirable owing to the possibility of leakage of hot dust-laden gases into the boiler room. Usually a stack is provided of sufficient height to handle the gases through breeching and connections and maintain a slight negative pressure at the fan outlet. Unless a separate stack serves the boiler unit being considered, it is unwise to credit the stack with much draft in selecting an induced-draft fan. With several older boilers connected to the same stack, it will be difficult to predict the operating conditions of all these boilers in order to determine accurately the total weight and mean temperature of gases in the stack. It is usually safe, however, to assume that no positive pressure will be required at the fan outlet and that the static

pressure required will be the sum of the resistances between furnace and fan inlet plus a reasonable allowance for furnace draft.

Owing to the presence of air infiltration, the same weight of gas does not pass through each piece of apparatus so that velocities and resistances for boiler, economizers and air heaters must be calculated independently, corrected in each case for estimated air infiltration.

In addition to these losses, the fan inlet boxes must be taken into consideration. It is best, however, to purchase these as a part of the fan unit and to specify that the required pressure is exclusive of inlet box losses since this item will then be allowed for by the manufacturer in the design of the fan. But even if these precautions are taken, an improper duct arrangement ahead of the fan inlet or beyond the discharge may affect the fan performance and it is wise to submit the duct arrangement to the fan manufacturer for comments.

If the calculated actual pressure requirements are based on gas weights considered obtainable only under favorable operating conditions, it is well to specify at least 30 per cent excess pressure in selecting the fan.

TABLE II—CHECK LIST OF ITEMS FOR CALCULATION OF INDUCED-DRAFT FAN REQUIREMENTS

		Volume
1.	Excess Air	Is it conservatively specified or absolute minimum?
2.	Infiltration	Are allowances conservative or minimum?
3.	Leakage	Allow for air to gas leakage in regenerative air heater.
4.	Moisture	Allow for wet fuel, hydrogen, combustion air, humidity, ashpit jets.
5.	Dust Collector	Does collector require recirculation of gas?
6.	Temperature	Is temperature used conservative or minimum?
		Temperature and Density
7.	Variation	Dirty heating surface and lower excess air will raise temperature.
8.	Bypass	When used, how will temperature be affected?
9.	Moisture	Affects density.
10.	Fly Ash	Affects density.
11.	Elevation	Lower density air increases volume and system resistance, and decreases fan performance.
		Pressure
12.	Outlet	Will fan be required to assist stack?
13.	Stack	Can stack be depended upon to assist fan?
14.	Inlet	Add all series resistances; are they conservative or minimum?
15.	Evasé	Properly proportioned outlet will recover velocity pressure.
16.	Elevation	Has "stack effect" in all apparatus and ducts been allowed for?
17.	Dust Collector	Has one been allowed for or specified for future installation?

Primary Air Fans

These fans are used only with a pulverized coal storage (or indirect) system. In this system, the air used in the pulverizer for drying and handling the coal is separated from the coal in the collector. At or near the feeder, coal and air (known as primary air) are again mixed together in proper proportions for ignition at the burners.

They may be classified according to the source of the air handled; *first*, installations where the primary air is taken from the room, preheated air duct or both and *second*, those installations where the vented air from the pulverizer system is used as all or part of the primary air.

On installations of the first classification, only the air necessary for proper carriage of the coal to the burners, and to give the required velocity to the coal-air mixture at the burner is used. This quantity depends, therefore, on the type of burner employed, the kind of coal and, in some cases, also the burner and furnace arrangement. A considerable amount of control of flame and furnace conditions can be accomplished by varying the primary air quantity and the discharge velocity at the burners.

On installations of the second classification, vented air is available only when the pulverizers are operating, which, with the storage system, may not be all the time that the boiler is in operation. At other times, arrangement must be made for the use of preheated air or cold air, or both, to be handled by the same fan. The vented air available depends on the size of the pulverizers, the moisture removal from the coal, and upon the temperature and quantity of air used for drying. For a given set of these conditions, the vent will be constant in quantity and cannot be varied to suit burner requirements. Therefore, the pulverizers cannot operate at full capacity when the boilers are operating at low capacity unless the burners can handle all the vented air or an alternate venting system is provided. At high boiler ratings if the vented air is not sufficient for proper operation of the burners, arrangements must be made to supply the deficiency as preheated or cold primary makeup.

The primary air temperature that can be used at the feeder depends on feeder and feed pipe arrangement. Temperatures as high as 500 F have been used under proper conditions. For best overall efficiency, primary air should be used without tempering or the quantity should be kept at a minimum so as to take advantage of the full preheat of a larger amount of secondary air.

The temperature of the air that the fan is to handle depends on the location of the fan. If preheated air is to be used without tempering, two arrangements may be used; (1) a separate preheater for primary air may be used with the primary air fan handling cold air; or (2) the fan may receive preheated air from the hot air duct of the large preheater and discharge it to a primary air header at the required high pressure.

When full preheat cannot be used, the air is generally tempered at the suction side of the fan by dampering the preheated air to the fan and placing an adjustable cold air opening between this damper and the fan inlet. The temperature of the air handled by the fan with such an arrangement should be the maximum that can be used with the particular feeder and piping arrangement being used.

When vented air is to be used at such times as it is available, the temperature will depend on the vent temperature, the location of the vent on the return air line, and on whether or not preheated or cold air is added to the primary air at the fan inlet. The vented air will usually be of 70 to 100 per cent relative humidity, between 110 and 150 F, and containing, of course, considerable pulverized coal.

Generally speaking, the quantity of primary air required may be estimated with a greater degree of accuracy than the secondary air or flue gas since it is not necessarily affected by changes in excess air supplied to the furnace. It is, however, affected to some extent by the design of the furnace since primary air velocity is a factor in controlling flame shape, particularly flame length. For this reason, definite limits cannot be set for a given burner for its entire operating range because so many factors are involved including the skill of the operator. The maximum requirement at maximum burner capacity can, however, be quite accurately calculated for use in estimating fan volume.

It will generally be sufficient to allow 10 per cent excess volume above calculated requirements in selecting primary air fans.

The fan pressure depends on the arrangement of fan and ducts and upon the source and temperature of the

air handled. When the fan handles cold air delivering it direct to the feeder, or first to an air heater, the pressure at the fan inlet will be atmospheric and the total static pressure will be the sum of the resistances between the fan and the furnace, including allowance for acceleration at the burner. When the fan handles preheated air delivered under pressure direct from the hot air duct, then the pressure available at the fan inlet may be subtracted from the total static pressure requirements calculated between fan and furnace, except when this preheated air must be tempered. When the temperature is too high for direct use, the available positive pressure is usually dissipated by dampering and tempering air admitted by operation with a negative pressure of -0.5 to -1.0 in. w.g. at the fan inlet.

When the primary air fan handles the vented air, the pressure required at the inlet will depend on the venting arrangement. When no vent fan is used, the primary air fan will be required to draw the air from the return air line which may be under considerable negative pressure.

Since the velocities in primary air piping are high, the effect of changes in velocity cannot be neglected in these calculations.

If the pressure required to accelerate the coal at the feeder has been determined by test, it should be possible quite accurately to calculate the other losses and 10 per cent excess pressure should be sufficient for fan selection.

Vented-Air Fans

It is customary to supply preheated air or hot flue gas to accomplish drying of the coal in a pulverizer in order to increase the pulverizer capacity. With the storage system, it is advantageous to remove the moisture from the coal in order to reduce difficulties in handling, storing and feeding the pulverized coal. The drying medium supplies part of the heat to vaporize the moisture and at the same time approaches saturation, making it of no further use for drying in the pulverizer system. A quantity of this nearly saturated mixture approximately equal to the weight of the hot drying medium supplied must be vented from the system. The amount vented, therefore, depends on the pulverizer output, the moisture removed from the coal, the temperature of the drying medium and the efficiency of the drying system. The latter may be measured in terms of relative humidity. The vented mixture may be disposed of in two ways; (1) by venting to the atmosphere or the stack, and (2) by using it as a part of the air supply to the burners or furnace. In the first case, since the vent contains considerable coal, it cannot be vented direct without first removing most of the coal. This may be done in a cyclone concentrator or in an air washer or both. In the second case, the air need not be cleaned, but in both cases there is sufficient resistance in the venting system to require a separate fan unless in the second case, the primary air fan is designed for additional pressure to handle the air.

When vent fans are used in connection with systems venting to the secondary air at the burner, the fan must handle a quantity equal to the weight of hot air supplied for mill drying plus the mill leakage and moisture removed from the coal.

Vent fans, therefore, are called upon to handle dirty, nearly saturated air. Vented air piping must be designed for high velocities and without pockets or horizontal runs where coal might collect and cause fires.

In specifying volume for the selection of vent fans, the effect of the presence of coal and moisture on the density of the mixture should be considered and the temperature and density should both be given to the fan manufacturer. Allowance should be made for the possible use of more air for drying in the pulverizer than theoretically required due to lower available temperature or higher moisture of the coal, and for the air that is used to "blow" the concentrator when such an arrangement is used. For fan selection, approximately 20 per cent excess volume is advisable.

On the inlet side of a vent fan, allowance must be made for the pressure at the point where the vent is taken off and for the resistance of the inlet line. On the discharge side, the vent line itself may represent a large item but the other items must also be considered depending on where the air is vented, namely: concentrator or air washer pressure drop, burner windbox pressure and furnace pressure. The concentrator pressure drop will, of course, depend on the design of the concentrator and seemingly slight changes in its proportions will greatly affect efficiency and pressure drop. Test results on an efficient design should be used in setting up this loss.

So far as excess pressure recommendations are concerned, it may be stated that if there is a possibility that the fan might be called upon to handle the above mentioned 20 per cent excess volume, then it would be well to specify 40 per cent excess pressure.

Overfire Air Fans

As stated in the introductory remarks, these fans are used primarily with chain grate stokers burning bituminous coal, and in the discussion on forced-draft fans, the sources of the air used were mentioned. The function of overfire air is to create turbulence in the furnace at the proper zone in order to prevent stratification of gases. It, therefore, aids in the elimination of smoke and in the combustion of the volatile in the coal and the carbon monoxide produced by the fuel bed. With a rear-arch furnace and a long stoker, one of the functions of the arch is to prevent the lean gases at the ashpit end of the stoker from passing direct to the upper part of the furnace. By forcing the gases to flow in a nearly horizontal direction toward the front, the rich and lean gases are mixed below the throat. When operating at reduced load with a shorter fire, this mixing action is less effective and overfire air is of great assistance. It is apparent, therefore, that a knowledge of stoker operation and furnace design is necessary in order to determine the necessity for and proper application of overfire air. In general, it may be stated that the jets should be large enough and so designed as to give proper penetration. Two or two and a half inch diameter openings spaced 12 in. apart and discharging at 100 to 150 ft per sec velocity have been found satisfactory in most instances. The total nozzle area and desired velocity will determine the quantity to be handled by the fan and this will run between 10 and 15 per cent of the total combustion air. The discharge velocity and design of ducts, dampers and plenum chambers will determine the pressure required. This will run between 3.5 and 7.0 in. w.g. in the plenum chamber. Little or no excess volume and pressure need be allowed because the capacity of this fan

will seldom limit the maximum capacity of the steam generating unit.

Pulverizer Exhausters

Pulverizer exhausters handle the coal and air mixture through the mill. They handle a higher concentration of solid material than any other fan mentioned herein and operate against high pressures, and at very high tip speed. The air-to-coal ratio will vary at full capacity

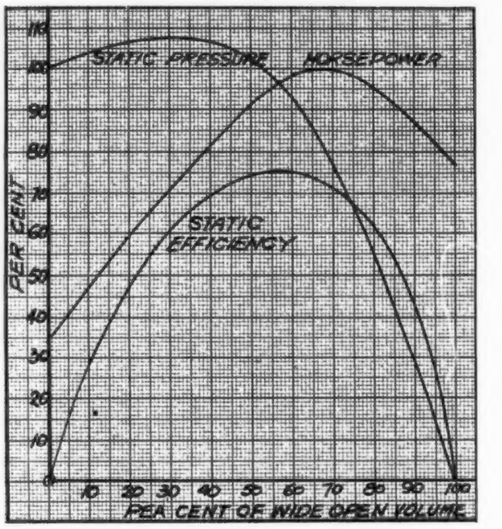

Fig. 1—Typical constant-speed characteristic curve for fan with backward-curved blades

from about 1.5/1 to 3/1 depending on the coal, the air temperature and the kind of pulverizer. The temperature will vary between about 125 F and 200 F although higher temperatures have been used with some pulverizers. For reasons of economy involving air heater performance, the minimum air flow consistent with design limitations should be used if the preheated air must be tempered. The air flow necessary for any given pulverizer can only be determined by test; each type will have different characteristics. It will also vary with other factors such as coal grindability, moisture, desired fineness and with the rate of grinding and size of pulverizer.

So much for the factors affecting calculation of the required volume. The pressure against which exhausters must operate will depend on the system of firing. With direct firing pressure allowance must be made to draw air into the pulverizer, to effectively sweep the pulverized coal from the grinding section into the classifier and to discharge the air and properly classified coal through piping, distributors and burners into the furnace at proper velocity. There is, therefore, a close relationship between requirements of burner, pulverizer and exhauster.

With the storage system, the pressure requirements may be the same as above on the inlet side but on the discharge side, the separator or collector is used to separate the coal and air, the air being returned in part to the pulverizer inlet.

The pulverizer exhauster is furnished with the pulverizer either as an integral part driven by the same motor or as a separate unit. In either case, it is considered a part of the responsibility of the pulverizer manufacturer to select the proper equipment.

Characteristic Curves

Fig. 1 shows a typical set of constant speed characteristic curves for a fan having backward-curved blades. The performance of a single fan of a given type can be used to determine the characteristics of a complete line of sizes by application of the theory of similitude, but each separate type must be tested independently. The curves shown in Fig. 1 are drawn for one speed only and, therefore, when the fan is driven by a constant-speed motor, it will operate somewhere on the characteristic curves depending on the resistance imposed by the system through which the air is being passed. The system resistance, therefore, acts as an automatic control on the fan and will limit the amount of overload that the fan can cause regardless of the flatness of the pressure characteristic. The amount of overload will depend also on the shape of the power characteristic. The backward-curved blades give a non-overloading characteristic but all other types have increasing power right up to wide open volume. The power required at wide open volume for the radial-tip, straight radial-blade and forward-curved blade will be, respectively, approximately two times, two and a half times, and three times the power required at the point of maximum static efficiency.

The characteristic curves for any other speed and/or air density may be constructed by application of the laws of fan performance.

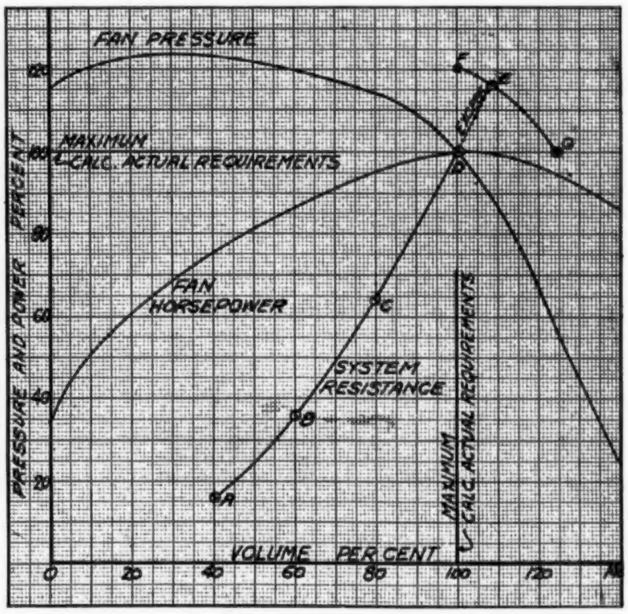

Fig. 2—Illustrating application of characteristic curves to fan problem

Fig. 2 illustrates the application of the characteristic curves to a fan problem where points *A*, *B*, *C* and *D* represent calculated requirements at four loads on a boiler. The line through them defines the system resistance and the point where this line intersects the static pressure characteristic of any fan, at any speed, will determine the point on the characteristic at which that fan will operate, if both curves are plotted for the same density. However, the fan can only operate on its characteristic curve so that if any error has been made in the calculation of point *D*, i.e., either volume, pressure or temperature, it will not fall on the characteristic curve and the fan may

not satisfy the requirements, operating at that particular speed. For example, if 10 per cent more volume is needed at the same pressure point, D will be displaced to the right but the available pressure at the fan has at the same time dropped 14 per cent and the fan cannot satisfy the requirements. Similarly, if the volume were correct but 10 per cent more pressure were needed, the volume that the fan would deliver at this increased pressure would be only about 90 per cent of the requirements since the fan can only operate on its characteristic curve for a given speed and density.

To obtain excess capacity or, in other words, a larger fan, it is customary to specify volume and pressure in excess of the actual calculated requirements. Suppose a portion of the pressure characteristic of a larger fan operating at the same speed and density is represented by the line FG, then it is obvious that this size fan would be selected by the manufacturer if the purchaser specified 24 per cent excess volume with no excess pressure or 20 per cent excess pressure with no excess volume, and the fan would at the same time satisfy the requirements of point E which requires 8 per cent excess volume and 17 per cent excess pressure. The only advantage in attempting to define point E on the extrapolated system resistance curve, instead of point F or G, is that the power requirements given by the manufacturer will then represent a closer estimate for the larger fan under actual operating conditions, than if points F or G had been defined for fan selection. The fan finally selected will, however, be capable of satisfying the requirements of all three points if sufficient power is available from the drive.

Fan Laws

1. *For a given fan size, piping system and air density:*

A When speed varies

(a) Capacity varies directly as the speed ratio.

(b) Pressure varies as the square of the speed ratio.

(c) Horsepower varies as the cube of the speed ratio.

B When pressure varies

(a) Capacity and speed vary as the square root of the pressure.

(b) Horsepower varies as the $(\text{pressure})^{1.5}$.

2. *For constant pressure:*

When density varies, speed, capacity and horsepower vary inversely as the square root of the density; that is, inversely as the square root of the barometric pressure and directly as the square root of the absolute temperature.

3. *For constant capacity and speed:*

When density of air varies, horsepower and pressure vary directly as the air density; that is, directly as the barometric pressure and inversely as the absolute temperature.

4. *For constant amount by weight:*

A When density of air varies

(a) Capacity, speed and pressure vary inversely as the density; that is, inversely as the barometric pressure and directly as the absolute temperature.

(b) Horsepower varies inversely as the square of the density; that is, inversely as the square of the barometric pressure and directly as the square of the absolute temperature.

B When both temperature and pressure vary

(a) Capacity and speed vary as the square root of (pressure $\times$ absolute temperature).

(b) Horsepower varies as the square root of ($\text{pressure}^3 \times$ absolute temperature).

Fan Types

Most of the mechanical draft fans used in power plants are of the centrifugal type. These employ blades mounted on an impeller rotating within a spiral or volute housing. Blade design determines the characteristic. A velocity vector diagram at blade tip will indicate that backward-curved blades produce low resultant velocities for a given tip or peripheral speed and forward-curved blades give high velocities. Radial blades and radial-tipped blades lie between these two extremes. The backward-curved blade type, therefore, operates at higher motor speeds than the other types for a given duty, and is well adapted to direct drive with motors or steam turbines.

There is a reason and a field for each type and the selection is best left to the manufacturer who will guarantee satisfactory operation. Besides the operating conditions of volume, pressure and temperature specified by the purchaser, many other factors must be considered such as, method of drive, speed of drive, method of control, space limitations, allowable noise and vibration, corrosion and/or erosion due to materials contaminating the gases, range of operation, point of most continuous operation, etc.

Direct-connected drives are used almost exclusively in power plant work. Control is obtained by the use of variable-speed motors, hydraulic couplings, specially designed inlet dampers, or plain louvre dampers. Forced-draft fans are generally of the backward-curved blade type and, when used with 60-cycle a-c drives, are operated at motor speeds of 1800 or 1200 rpm. Induced-draft fans for moderate pressures are generally of the forward-curved blade type and will operate generally at motor speeds of 720 or 900 rpm but never exceeding 1200 rpm. For higher pressures, straight blade or radial tip fans are sometimes offered, operating at the same speeds as the above. Vent fans, primary air fans and pulverizer exhausters are invariably of the straight-blade type and operated up to 1800 rpm, depending on the pressure requirements. Fans having backward-curved blades are sometimes offered for induced-draft service with gas or oil firing because they permit use of higher motor speed and give the highest efficiency of any centrifugal type. However, if there is any possibility of dirt in the gases, it may deposit on the back of the blades and give trouble.

In concluding this discussion on the factors influencing the requirements and recommended excess factors for fan selection, it might not be amiss to consider the importance of the fans of a steam generation unit in their relation to maximum generating capacity. Many units representing a large investment would be capable of generating more than their rated steam capacity if it were not for fan limitations and many other units have fallen shy of rated capacity because fans were figured too closely. There is always the chance that the investment of a few extra dollars in the fans may make available considerable increased capacity, and the certainty that with ample capacity in the fans, the operators need not fear the sudden occurrence of heavy load demands even when conditions of operation are sub-normal.

NEW CATALOGS AND BULLETINS

Any of these publications will be sent on request

Boiler Water Conditioning

The Elgin Softener Corporation has published a new edition of its booklet entitled, "The Inside Story of Boiler Water Conditioning" which discusses variations in water and in operating conditions, how impurities accumulate inside a boiler, dissolved impurities in boiler feedwater and the essentials of boiler-water conditioning. The Elgin system is described.

Combustion Control

A new 16-page bulletin on Automatic Combustion Control has just been issued by the Hagan Corporation. This discusses at length the subject of combustion control, describes the Hagan system and gives layouts of typical installations as applied to various types of boilers and for different fuels. It is attractively printed and well illustrated.

Economizers

Combustion Engineering Company, Inc., New York, has issued a new 16-page catalog dealing with the latest designs and details of Elesco fin-tube economizers. These are of two general designs—Type A in which the ends of the bifurcated tubes are connected by bends with flanged connections, and Type C in which forged return bends are employed. The former type is for use where feedwater conditions make ready access for internal inspection or cleaning desirable and the latter is for use where internal cleaning is not required. The extended surface on both types promotes high efficiency due to the high ratio of gas surface to water surface. The design is such as to provide low draft loss in relation to gas flow and high heat recovery in relation to draft loss. Also, due to the maintenance of effective distribution and velocity of gases at all rates of output and to the absence of eddy currents, soot and dust accumulations are reduced to a minimum.

Besides fully describing and illustrating these designs in detail, the catalog discusses factors governing the selection and application of economizers and shows numerous typical installations in modern steam generating units.

Flexible Couplings

The Poole Foundry & Machine Company has just issued a most attractive and informative 70-page catalog on its line of flexible couplings and their application to drives for a wide variety of both industrial and power plant equipment. For each type tables are given containing dimensions, speeds, weights and capacities. The catalog is fully illustrated and is provided with a convenient ring binding.

Fly Ash

A 16-page catalog has lately been issued by Buell Engineering Company describing the Van Tongeren System and the Buell fly ash collector the design of which is based on that system. Performance curves are included and photographs of typical installations, both here and abroad, are shown. This collector is built in two types, namely, the simple cyclone type adapted for use where the volume of flue gas to be cleaned is relatively small and the compound type for handling larger volumes of gas. The catalog is fully illustrated.

Hydraulic Coupling

The Hydraulic Coupling Division of American Blower Corporation has a new bulletin devoted to its traction type hydraulic coupling for use in connection with electric motor and internal-combustion drives. When used in connection with alternating-current motors this fluid coupling renders normal torque motors suitable for starting heavy loads, and when motors are operated in parallel it divides the load equally. The construction and operation of the coupling are described and illustrated and numerous applications are shown.

Power Plant Equipment

Yarnall-Waring Company has just issued a new revision of its 12-page bulletin, G-1304, covering in condensed form its line of steam plant equipment. This deals with Yarway blow-off valves of various types, water columns, expansion joints, spray nozzles and steam traps.

Steam Jet Ejectors

Ingersoll-Rand Company has recently issued a new bulletin covering single-, two- and three-stage steam jet ejectors for removing air, gas or vapors from condensers and vacuum chambers in industrial processes. Features of this type of vacuum pump include the ability to handle any quantity of wet or dry mixtures at any vacuum, high sustained efficiency, low initial and maintenance cost and economical trouble-free operation.

This 28-page bulletin explains the application and characteristics of ejectors and illustrates the operation and arrangement of all types of ejectors with either surface or barometric precoolers, inter- and after-condensers.

Variable-Speed Control

A 124-page catalog (G-384) has just been issued by the Reeves Pulley Company covering its complete line of variable-speed control equipment. This includes variable-speed transmissions, variable-speed motor pulleys and moto-drive. The operating principles are explained and fully illustrated as are also all details including automatic controls, lubrication, etc. Over fifty pages are devoted to engineering data. There are more than 200 illustrations many of which show typical applications. The book is printed in two colors and is bound in a heavy embossed cover with a convenient ring binding.

Testing Boiler Waters for Embrittlement*

By F. G. STRAUB[1] and T. A. BRADBURY[2]

The paper describes a new method of testing with actual samples of boiler water to determine whether they will cause embrittlement. A large number of such tests indicates that the presence of sodium chloride aids sodium sulphate in preventing embrittlement at pressures up to 350 lb. At pressures above this sodium sulphate has practically no inhibiting action, but small amounts of aluminate may be effective in this range.

THE problem of the cause of embrittlement in steam boilers and methods of its prevention has been very thoroughly studied since the early days of the occurrence of this insidious type of steam boiler distress. Much research work has been conducted in order to answer the various questions which have been asked in regard to this type of boiler failure and these researches have been quite fruitful in their results. However, the boiler operator is still faced with the problem of trying to determine whether his particular boiler water is embrittling in nature and if it is, what he has to do to make it non-embrittling.

It was with the purpose of answering these questions for the boiler operator that the work herein described was started. Early in the work, it was realized that there was need of a simple method of testing, whereby a sample of boiler water could be tested in such a manner as actually to produce embrittlement in a piece of boiler steel under conditions directly comparable to boiler operation. Embrittlement has been reproduced in boiler steel but under conditions which made various assumptions necessary. First, the concentrations of the chemical involved were about one thousand times that existing in the boiler; second, the steel was tested in a container so that a large volume of concentrated solution surrounded the test specimens; and third, actual boiler waters could not be directly tested under such conditions.

A survey of the instances of embrittlement found in steam boilers showed that they always occurred in an area where a capillary space existed and where concentrated stresses might occur. Leaks to the outside were apparently not necessary.

Iron in contact with sodium hydroxide tends to react and form the iron oxide and liberate hydrogen at relatively low concentrations of sodium hydroxide and at boiler water temperatures. If this concentration were to take place in a boiler seam where a small volume of water is in contact with a relatively large area of iron, the reaction of a partially concentrated sodium hydroxide solution with the iron to evolve hydrogen would remove the water and concentrate the caustic, thus causing the reaction to proceed further. In this manner a very concentrated caustic solution could be produced without the temperature becoming high enough to boil the water out of the solution. The testing of a concentrated solution surrounding a test specimen where free circulation is possible precludes the possibility of this second reaction taking place. Furthermore, the concentration within a seam would produce other conditions which would not be reproducible under the older methods of testing.

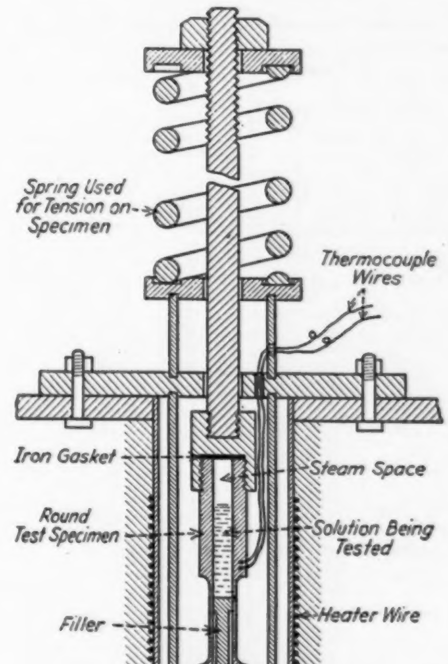

Fig. 1—Section of testing unit

To take advantage of these ideas, the method of testing should involve the placing of a capillary space adjacent to the steel under tension. The water to be tested should have access to this capillary space, and free circulation must be prevented. Fig. 1 shows a section of a test unit devised to test dilute or concentrated solutions. The

* Abstracted from a paper before the Annual Meeting of the American Society for Testing Materials at Atlantic City, June 27–July 1, 1938. The investigations here reported were conducted under a cooperative agreement between the Utilities Research Commission, Inc., and the Engineering Experiment Station, University of Illinois.

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test specimen is made of a known steel. In these tests, S.A.E. No. 1020 hot-rolled steel has been used, although any type of steel might be used. The steel specimen is 1 in. round and about 5 in. long. A hole 0.5 in. in diameter is bored into the specimen and a portion of the steel removed near the bottom so as to give a reduced section. When this specimen without the filler was partially filled with a solution of caustic soda containing silica, assembled in the test unit, heated and stressed by compressing the spring, it was found necessary to have concentrated solutions of caustic present before failure resulted. This was as expected, since this method of testing is similar to the older ones used except that the solution is inside the specimen and thus eliminates packing glands along with potential points of leakage.

The method was then modified by inserting the small solid steel filler inside the test specimen. The filler is made a few thousandths of an inch smaller than the inside of the specimen at the top and reduced still smaller at the bottom. This allows a capillary space to exist in contact with the stressed area of the specimen. This design of filler was used so as to allow the test solution to have access to the capillary space where it could concentrate, in contact with the stressed steel, without removing steam from the test specimen. This is very similar to the stressed areas in the boilers where embrittlement takes place. When dilute solutions of caustic soda were tested in this unit, failure resulted.

TABLE I—EFFECT OF TEMPERATURE AND TYPE OF LOADING ON HOLLOW SPECIMEN WITH FILLER.

Solution used contained 800 p.p.m. NaOH 80 p.p.m. SiO_2

Test	Temperature, deg. Fahr.	Load Applied	Load Applied, lb. per sq. in.	Results of Tests	
				Break	No Break
No. 1710.....	300	Hot	40 000	2 hr.	
No. 1610.....	400	Cold	45 000	....	25 days
No. 1617.....	400	Cold	50 000	2 days	
No. 1619.....	400	Cold	55 000	10 days	
No. 1608.....	400	Cold	40 000	6 days no break; load hot 45 000, break 3 hr.	
No. 1612.....	400	Hot	40 000	2 hr.	
No. 1618.....	400	Hot	40 000	2 hr.	
No. 1648.....	400	Hot	40 000	2 hr.	
No. 1686.....	400	Hot	35 000	5 hr.	
No. 1689.....	400	Hot	30 000	19 hr.	
No. 1603.....	500	Cold	40 000	2 to 4 hr.	
No. 1604.....	500	Cold	35 000	4 to 15 hr.	
No. 1605.....	500	Cold	30 000	6 days no break; load hot 35 000, break 1 hr.	
No. 1606.....	500	Cold	35 000	0 to 10 hr.	
No. 1685.....	500	Hot	30 000	breaks	
No. 1687.....	600	Hot	35 000	0 to 10 hr.	
No. 1688.....	600	Hot	30 000	$1\frac{1}{2}$ hr.	
No. 1692.....	600	Hot	25 000	....	10 days

Table I gives the results of tests run at temperatures between 300 and 600 F using a solution containing 800 ppm of NaOH and 80 ppm of SiO_2. The solution used in the hollow specimen tests without the filler, in which no failure resulted, was approximately 50,000 ppm or 60 times the concentration of the dilute solution which caused failure when the filler was used. As shown in Table I, two methods of loading were used, one where the load was applied at room temperature and the other where the specimen was heated to the test temperature, held there for four or five hours, and then subjected to the desired stress. These results show that apparently the insertion of the filler has brought about a condition whereby dilute solutions will concentrate in the capillary space and in the presence of sufficient stress produce failure of the steel. In test No. 1692 where no failure resulted, the solution above the filler at the end of the test had only 60 ppm of sodium hydroxide in solution. This showed a reduction from 800 to 60 ppm. The sodium hydroxide had been concentrated in the capillary space with the reduction of concentration in the solution above. The removal of 740 ppm of NaOH from the top solution and concentrating in the lower capillary would increase this concentration to 10,000 ppm as a minimum value.

In order to check this method of testing, boiler water samples were collected from power plants where embrittlement was being experienced as well as from plants not having any trouble. More than 200 such boiler waters[3] have been tested at temperatures between 360 and 600 F (150 to 1500 lb steam pressure). By assembling a representative group of tests conducted at the various temperatures, it is possible to illustrate the action of the various salts in preventing embrittlement. In order to do this, it was found advisable to assemble the results under the following temperatures:

(1) 360 and 400 F (150 and 250 lb pressure)
(2) 470 and 570 F (500 and 1400 lb pressure)
(3) 425 F (350 lb pressure)

Tests Conducted at 360 and 400 F:

Tables II and III give a typical group of tests conducted at 360 and 400 F in which no phosphate occurs. Only a few tests were conducted at 360 F; however, a definite trend is noticeable in the results obtained. These are (1) failure results even with an A.S.M.E. ratio of 3.0, (2) sodium chloride aids materially in preventing failure. Thus in boiler water No. 1066 increasing the ratio of the sodium chloride to the total alkalinity from 0.06 to 0.40 changed the time of failure from less than 11 hr to 29 days when the A.S.M.E. ratio was only 0.51. When boiler water No. 1068 was tested, failure took place even with an A.S.M.E. ratio of 3.13. When the chloride content was increased so that it was 0.27 times the alkalinity, no failure resulted. In boiler water No. 1088, no failure resulted with a ratio of sodium chloride to alkalinity of 0.40 and an A.S.M.E. ratio of 0.90.

Table III gives the results of a representative group of tests conducted at 400 F. Here it is evident that neither the sodium sulfate nor the sodium chloride alone prevents embrittlement. However, if proper amounts of both salts are present, embrittlement may be prevented. Tests Nos. 1700, 1669, 1703 and 1715 illustrate this very clearly. When the ratio of the sodium chloride to the total alkalinity is plotted against the A.S.M.E. ratio, it becomes evident that there is apparently a minimum value for the sodium chloride–alkalinity ratio (0.60) above which embrittlement is stopped even though the A.S.M.E. ratio is as low as 1.0. If this sodium chloride–alkalinity ratio is reduced below 0.60, embrittlement results even though the A.S.M.E. ratio is high. There ap-

[3] These waters were analyzed as follows: Total alkalinity, chloride and sulphate (gravimetric) by A. P. H. A. method. Silica and R_2O_3 gravimetrically, using perchloric acid for the silica.

pear to be instances where lower chloride–alkalinity ratios are effective. This may be due to the fact that the effect of carbonate, undetermined organic, nitrates, etc., have not been considered.

It is evident that in waters which have caused cracking and where the sodium chloride–alkalinity ratio has been below 0.60, increase in the sulphate content has not prevented failure, while in waters which have caused cracking and which have the A.S.M.E. ratio around 1, increasing the sodium chloride–alkalinity ratio to 0.60 or greater has stopped cracking.

Tests were run at 400 F using boiler waters having phosphate present. It does not appear that sufficient data are available to state definitely that phosphate does or does not prevent embrittlement. It may be stated that phosphate alone does not prevent failure. It appears possible that the presence of chloride might have a marked effect on the phosphate action.

Tests at 470 and 570 F.:

A group of tests conducted at 470 F showed that the chloride–sulphate effect so noticeable at 400 F apparently disappeared at this temperature. However, embrittlement is prevented at this pressure and it appears to be due to the R_2O_3 content. The R_2O_3 content, being a combination of iron and aluminum, is a rather indefinite value. However, in the majority of the waters tested, the Al_2O_3 appeared to predominate. Tests are being conducted to study the relative effect of the iron and aluminum.

TABLE II.—RESULTS OF ANALYSES OF WATERS AND OF TESTS CONDUCTED AT 360 F. (150 LB. PRESSURE).

Test	Water	Total Alkalinity, p.p.m.	Ratio, NaCl to Total Alkalinity	A.S.M.E. Ratio, Na_2SO_4 to Total Alkalinity	SiO_2, p.p.m.	Load Applied, lb. per sq. in.	Results of Tests	
							Break	No Break
No. 1759......	No. 1066....	1 410	0.06	0.51	72	45 000	0 to 11 hr.	
No. 1760......	No. 1066[a]..	1 410	0.40	0.51	72	45 000	29 days	
No. 1763......	No. 1068....	600	0.09	3.13	14	45 000	4 to 15 hr.	
No. 1766......	No. 1068....	600	0.09	3.13	14	40 000	9 days	
No. 1767......	No. 1068[a]...	600	0.27	3.13	14	45 000	....	26 days
No. 1828......	No. 1088....	10 150	0.40	0.90	395	45 000	....	29 days
No. 1852......	No. 1088[b]...	1 015	0.40	0.90	39	40 000	....	22 days

[a] Sodium chloride added.
[b] Diluted with distilled water.

TABLE III.—RESULTS OF ANALYSES OF WATERS AND TESTS CONDUCTED AT 400 F. (250 LB. PRESSURE).
No phosphate present.

Test	Water	Total Alkalinity, p.p.m.	Ratio, NaCl to Total Alkalinity	A.S.M.E. Ratio, Na_2SO_4 to Total Alkalinity	SiO_2, p.p.m.	R_2O_3, p.p.m.	Applied Load, lb. per sq. in.	Results of Tests	
								Break	No Break
No. 1620....	No. 21 630.	1 400	0.10	1.1	110	....	40 000	5 days	
No. 1623....	Synthetic..	1 400	0.10	1.1	110	0	40 000	2 days	
No. 1679....	No. 21 630[a]	1 400	0.10	2.2	110	....	40 000	4 days	
No. 1683....	Synthetic.	1 400	0.10	2.2	110	0	40 000	3 days	
No. 1718....	No. 21 630[b]	1 400	0.30	1.1	110	....	45 000	0 to 8 hr.	
No. 1735....	No. 21 630[b]	1 400	0.60	1.1	110	....	45 000	....	46 days
No. 1731....	No. 21 630[b]	1 400	5.00	1.1	110	....	45 000	....	27 days
No. 1706....	No. 1054...	1 430	1.10	9.0	124	....	45 000	....	54 days
No. 1725....	No. 1055...	795	0.25	0.4	90	....	40 000	0 to 11 hr.	
No. 1776....	No. 1071...	522	0.20	0.4	0	8	45 000	....	28 days
No. 1839....	No. 1080...	725	0.23	1.0	58	....	40 000	2 days	
No. 1905....	No. 1102	373	0.49	3.1	6	3	40 000	19 days	
No. 1954....	No. 1130...	502	0.96	2.9	9	9	40 000	....	34 days
No. 1975....	No. 1141...	316	1.13	3.1	38	5	40 000	....	30 days
No. 1976....	No. 1142...	304	1.17	3.8	25	4	40 000	....	30 days
No. 1659....	No. 1050...	1 180	0.06	1.4	180	....	40 000	0 to 12 hr.	
No. 1691....	No. 1050[b]..	1 180	0.35	1.4	180	....	40 000	17 days	
No. 1771....	No. 1069...	1 055	0.97	3.0	152	70	45 000	....	40 days
No. 1812....	No. 1084...	595	0.67	2.9	128	....	45 000	....	43 days
No. 1966....	No. 1129...	256	0.55	1.5	31	11	40 000	8 days	
No. 1965....	No. 1128...	305	0.50	1.3	21	24	40 000	7 days	
No. 1734....	No. 1061...	2 105	0.20	3.8	416	....	40 000	0 to 16 hr.	
No. 1669....	No. 1041...	1 432	0.74	0.9	230	....	45 000	....	70 days
No. 1703....	Synthetic..	1 400	0.75	0.0	230	0	45 000	1 hr.	
No. 1715....	Synthetic..	1 400	0.00	0.9	230	0	45 000	1 hr.	
No. 1700....	Synthetic..	1 400	0.75	0.9	230	0	45 000	....	54 days
No. 1953....	Synthetic..	520	0.34	0.2	2	3	40 000	0 to 16 hr.	
No. 1952....	Synthetic..	500	0.00	0.0	0	0	40 000	24 to 39 hr.	

[a] Na_2SO_4 added.
[b] NaCl added.

It has been shown that silica has a marked effect on the caustic action in producing embrittlement. It might be considered as catalyzing the reaction. If the silica could be combined with other chemicals so as to form a complex salt, it would no longer act as a catalyst and the embrittling effect of the caustic might be stopped. In considering the action of the chlorides and sulphates in preventing failure, it has been assumed that they prevent the action of the caustic soda after having been catalyzed. Thus, relatively large amounts of these salts are necessary. If, as appears possible at the higher temperatures, the silica effect might be neutralized, the amounts of chemical necessary would bear a direct proportion to the silica.

The tests run at 570 F and reported in Table VI[4] also show the effect of the R_2O_3 to SiO_2 ratio. In test No. 1993, failure resulted but the addition of 3 ppm of R_2O_3 as Al_2O_3 prevented cracking. In test No. 1964, the boiler water had no free sodium hydroxide ($BaCl_2$ test) and no failure resulted even though the R_2O_3–SiO_2 ratio was low. When 100 ppm of chemically pure NaOH was added, failure resulted. The addition of 56 ppm of Al_2O_3 along with the hydroxide prevented failure in test No. 2016.

Boiler water No. 1140, test No. 2055, with a very low silica content but with the R_2O_3 content equal to 2 ppm did not produce embrittlement even when sodium hydroxide was added. However, when 10 ppm of SiO_2 was added to the same solution and tested, failure resulted in less than 14 hr (test No. 2063). These tests show that the prevention of embrittlement at this temperature is definitely related to the silica and R_2O_3 contents.

This effect of the R_2O_3 does not appear to extend over

[4] Tables IV and V ommitted from this abstract.

the entire temperature range. The R_2O_3 appears to be effective at 470 F and not at 400 F, while the sulphate-chloride appears to be inactive at 470 F but active at 400 F. In order to determine the temperature at which these salts lose their inhibiting properties, tests were conducted at the intermediate temperature of 425 F.

Results of Tests of 425 F:

The results of a group of tests conducted at 425 F indicate that the chloride-sulphate effect in preventing embrittlement is active at temperatures up to 425 F but is not effective at 470 F. The R_2O_3–SiO_2 influence is not effective at 425 F but is active at 470 F and above.

Discussion of Results

The results of the tests conducted indicate that this method of testing boiler waters appears to give results directly applicable to boiler water treatment. Boiler water samples were obtained from seven power plants which have recently experienced embrittlement. All of these waters caused failure in the test units. Four of these waters met the A.S.M.E. ratios but had low chlorides. Two of the waters had phosphate present with low chlorides. By slight modification of these waters, they were made so as not to produce failure. Five of these plants have modified their water treatment so as to conform to the type found to prevent failure.

In several power plants operating in the 1400-lb pressure range after having had their boiler waters tested and found to be non-embrittling even with low A.S.M.E. ratios, it was decided to operate without maintaining the A.S.M.E. ratio.

The application of the results of these tests to the control of water treatment must be done conservatively and only after a sufficient number of tests have been conducted on the type of water under consideration. A test run with the new testing unit is a severe one. The conditions are such that they might not always be encountered in operation and if a water were to produce embrittlement in the test, it does not follow directly that the boiler using this water would be embrittled. However, if a water produces failure in these tests and a slight modification of the boiler water prevents failure, then it is safe to predict that the modified water would be an extremely safe type to use so as to be certain that the boilers will operate free from embrittlement.

The addition of aluminate to prevent embrittlement at the higher temperatures should only be started after careful consideration of all the factors involved. In certain installations, the use of small amounts of sodium aluminate appears logical while in others it might cause other troubles. Any reduction in the silica at the higher pressures would aid in raising the R_2O_3–SiO ratio and thus prevent embrittlement. In many of the higher pressure plants, it is being found that there is a natural balance of these two chemicals which makes the water non-embrittling in nature without any special treatment for the prevention of embrittlement.

Arrangements have been made whereby the Chemical Engineering Department of the University of Illinois will test boiler waters for any interested parties. These tests will involve analyses of the waters and testing samples in the new embrittlement testing units. A nominal charge will be made for this work to cover the cost of conducting the analyses and experiments. Any profit accruing as a result of these tests will be used by the department for further research on boiler water problems of general interest to the steam power plant operators. From time to time the results of these tests will be published.

TABLE VI.—RESULTS OF ANALYSES OF WATERS AND TESTS CONDUCTED AT 570 F. (1350 LB. PRESSURE).

Test	Water	Total Alkalinity, p.p.m.	Ratio, NaCl to Total Alkalinity	A.S.M.E. Ratio, Na_2SO_4 to Total Alkalinity	SiO_2, p.p.m.	R_2O_3, p.p.m.	PO_4, p.p.m.	Applied Load, lb. per sq. in.	Results of Tests: Break	Results of Tests: No break
No. 1815....	No. 1083[a]..	675	0.86	3.8	110	490	0	45 000	....	30 days
No. 1818....	Synthetic..	675	0.70	3.3	105	0	0	45 000	3 days	
No. 1910....	No. 1113...	520	0.34	0.2	2	3	0	40 000	....	28 days
No. 1993....	Synthetic..	520	0.00	0.0	2	0	0	40 000	0 to 5 hr.	
No. 1995....	Synthetic..	520	0.00	0.0	2	3	0	40 000	....	30 days
No. 1972....	Synthetic..	520	0.34	0.2	2	3	0	40 000	....	30 days
No. 1982....	No. 1153...	340	0.25	2.8	15	8	125	40 000	2 hr.	
No. 1964....	No. 1139...	181	0.29	0.0	18	6	54	40 000	....	30 days
No. 1997....	No. 1139[b]..	310	0.17	0.0	18	6	54	40 000	1 hr.	
No. 2016....	No. 1139[c]..	310	0.17	0.0	18	56	54	40 000	....	30 days
No. 1918....	No. 1107...	276	0.29	2.2	1	2	0	40 000	....	30 days
No. 1919....	No. 1108...	228	0.32	1.0	2	4	0	40 000	....	32 days
No. 1942....	No. 1109...	350	0.49	1.5	2	3	0	40 000	....	32 days
No. 1921....	No. 1112...	224	0.31	5.6	4	6	0	40 000	....	35 days
No. 1911....	No. 1114...	515	0.47	0.2	12	15	0	40 000	....	28 days
No. 1983....	No. 1154...	607	0.23	1.9	5	4	167	40 000	....	30 days
No. 1996....	No. 1162...	415	0.08	0.4	16	9	40	40 000	....	30 days
No. 1999....	No. 1163...	520	0.31	1.6	35	16	60	40 000	....	30 days
No. 2055....	No. 1140[c]..	310	0.01	0.0	0	2	57	40 000	....	30 days
No. 2063....	No. 1140[d]..	310	0.01	0.0	10	2	57	40 000	0 to 14 hr.	

[a] Test conducted at 600 F.
[b] NaOH added.
[c] Al_2O_3 and NaOH added.
[d] NaOH and Na_2SiO_3 added.

Conclusions

The conclusions reached from the data obtained so far may be summarized as follows:

1. A new method of testing samples of boiler water to determine whether or not they will cause embrittlement has been developed.

2. For steam pressures up to 250 lb per sq in. embrittlement may be prevented by maintaining the sodium-chloride content of the boiler water greater than 0.6 times the total alkalinity expressed as sodium carbonate along with the sodium sulphate content greater than 1.0 times the total alkalinity.

3. For steam pressures between 500 and 1400 lb per sq in., these results would indicate that the presence of a soluble R_2O_3 content of greater than 0.6 times the SiO_2 content of the boiler water prevents embrittlement.

4. For a steam pressure of 350 lb per sq in., the sulphate and chloride to alkalinity ratios appear to be effective in preventing embrittlement, although larger amounts may be necessary than at the lower pressures.

High-Pressure Steam Generators*

By HENRY KREISINGER

Combustion Engineering Company, Inc.

An analysis of the gains possible by adopting high steam pressures and temperatures, followed by a discussion of high-pressure steam generator design, feedwater heating, steam washing, superheater design and control, air heaters and furnace design for burning different fuels.

GENERALLY, any pressure above 400 lb is referred to as high steam pressure. Pressures up to 900 lb are called medium high pressures, and above 900 lb high pressures. A steam temperature of 850 F is considered conservatively high temperature and that of 900 F and above as high temperature. Medium high pressures are used in some central stations to avoid the necessity of high steam temperatures for low moisture in the expanded steam. In industrial plants medium high pressures are often employed to balance the power demand with the steam demand. At present a number of installations in central stations and industrial plants are in operation and others in course of construction with a steam pressure of 1200 lb and temperatures of 900 to 925 F at the turbine throttle.

High steam pressures were introduced about 12 years ago when two installations of 1200 lb pressure and 750 F total temperatures were made. The steam was expanded to 300 lb in a high-pressure turbine, reheated to 700 F and used in existing 300-lb condensing turbines. Both of these installations gave satisfactory results and were followed by others.

Reheating of steam after partial expansion complicates the design and operation of a plant. At present there is a decided tendency to do away with steam reheating and to use high initial steam temperatures that permit complete expansion of the steam to the desired condenser pressure.

Why High Pressure Steam Is Used

Heat cannot be utilized unless it is at sufficiently high temperature so that it will flow of its own accord to where it is needed. Heat in high-pressure steam is at higher temperature than that in low-pressure steam, and therefore more heat is available for useful work in the former. In the process of power generation high-steam pressure came into use because a greater percentage of the heat in the steam is available for generating power. In other words, the thermal efficiency of the high-pressure cycle is higher and less fuel is used to generate a required amount of power. The reduction in the fuel consumption is obtained at a small additional first cost of the high-pressure plant.

Fig. 1 shows graphically what can be accomplished by going to higher steam pressures. The five columns represent the heat distribution in the process of power generation with steam at three pressures and different temperatures. The principal items of the operating cycle are given at the top of each column. The height of the columns indicated by the scale at the left is equal to the heat in the fuel burned per kilowatt-hour net output. Each column is divided into five rectangles the heights of which represent the various quantities of heat lost or utilized. The top rectangle represents the boiler room losses which are the same in percentage of the total

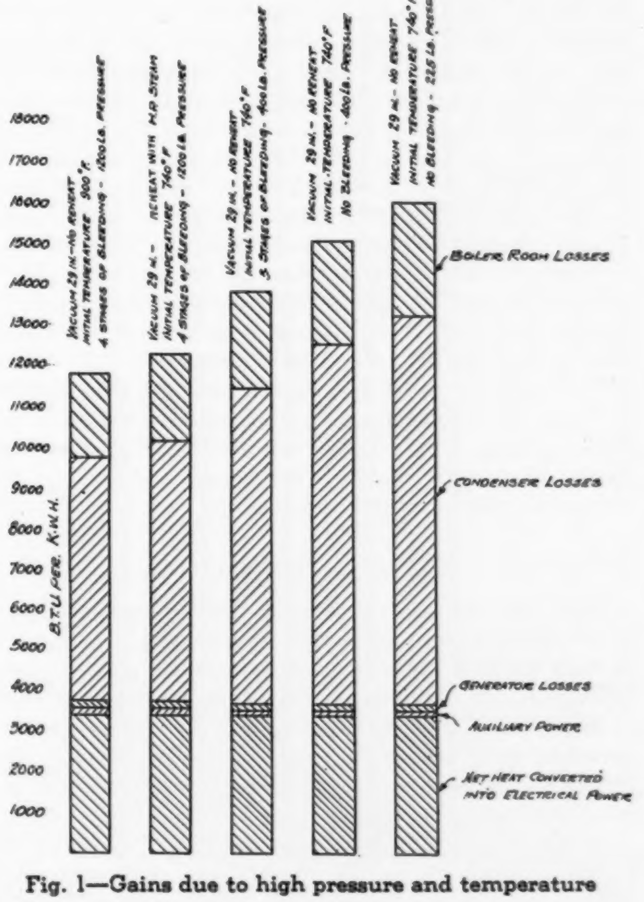


Fig. 1—Gains due to high pressure and temperature

heat used to generate one kilowatt-hour. The lowest rectangle represents the heat converted to net electrical energy of one kilowatt-hour and is equal to 3415 Btu.

The second and third rectangles from the bottom represent the energy used in auxiliary power and the generator losses, which quantities are the same in percentage of the net power generated for the 225- and 400-lb steam

* A paper delivered before the San Francisco Section, A. S. M. E., on May 5, 1938.

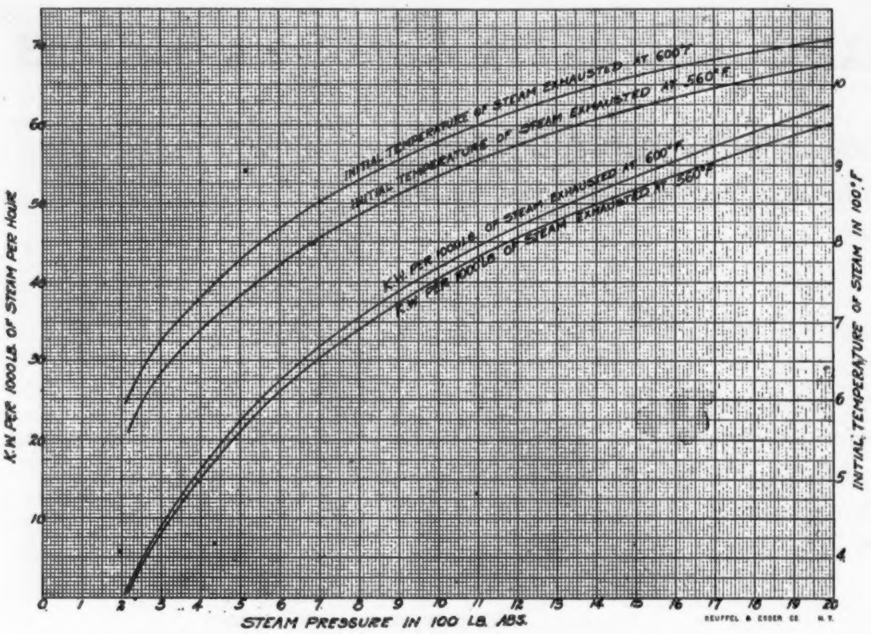


Fig. 2—Initial pressure and temperature combinations with exhaust at 560 and 600 F

alloys must be determined by laboratory tests. Although such tests are extensive they lack time factor of sufficient length. Because of the lack of sufficient operating experience with high temperatures, more conservative designers use medium high pressures with conservatively high steam temperatures.

Steam Reheating

Some of the high-pressure plants designed a few years ago used medium high initial temperatures with steam reheating after partial expansion. The reheating was done either with hot gases, the reheater being placed in the boiler setting, or, with high-pressure steam in which case the reheater was located near the turbine. Reheating with hot gases gives better thermal efficiency than reheating with high-pressure steam because there is some degradation of heat when the heat passes from the high-pressure steam to the partly expanded steam.

pressure cycles. With the two 1200-lb pressure cycles the energy used by the auxiliaries is somewhat greater because more energy is required to drive the feed pump. The fourth rectangle from the bottom represents the condenser loss which is the largest item in the process of generating power and the one which is greatly reduced by high steam pressures. The heat in the steam going to the condenser is at too low temperature, and therefore, not available for transformation into mechanical energy. It is an inherent loss of the steam cycle.

High-pressure steam also has the advantage in that a greater proportion can be bled or extracted from the turbine for feedwater heating after partial expansion. The condensate is at about 80 F and is heated by steam bled at suitable pressures in several steps to within about 150 deg F of the temperature of saturated steam at the boiler pressure. The water is heated by the latent heat of the bled steam which is condensed and thereby the condenser losses are further reduced.

High Initial Temperature

To avoid excessive moisture in the low-pressure stages of the turbine high-pressure steam is highly superheated so that most of the expansion takes place in the superheated region. Steam expanded from an initial pressure of 1200 lb and a temperature of 900 F contains 10 to 12 per cent moisture at the end of the expansion. The higher the efficiency of expansion the wetter is the expanded steam.

It is the high temperature of the steam that presents intricate problems in the design of superheater, stop valves and the steam turbines. One installation with 900-F steam has been in operation over two years without any difficulty and there is a strong urge to go to still higher temperatures. In fact, installations have been made for 925 and 950 F. More operating experience with high temperatures is needed to guide the designers of high temperature equipment. Alloy steel must be used in the construction of such equipment that is subjected to high temperatures and the properties of the

High-Pressure Plant Smaller

New stations are generally designed for high steam pressure and high temperature. The cost per kilowatt capacity of the high-pressure plant is only slightly greater than that of a low-pressure plant, because the former is smaller in size. Referring to Fig. 1 the size of the plant is proportional to the height of the columns. The steam-generating and the coal-preparation equipment becomes smaller as the pressure is raised, because there is less coal handled and burned. The condenser and its accessories are also much smaller because less steam is condensed. The fittings and piping for high-pressure steam are now standardized and are available on the market, without having to be designed specially. The use of welded construction of boiler drums eliminates the more expensive forged drums. Thus high-pressure and high-temperature plants provide means to generate power more efficiently at slightly greater first cost. With a reasonably good load factor it is hard to justify the economics of a low-pressure plant.

Modernization of Old Plants

By the superimposition of the high-pressure and high-temperature cycle on existing low-pressure equipment an out-of-date plant can be modernized at a reasonable cost for additional capacity. Such modernization requires new steam generating equipment and a high-pressure turbine exhausting directly to the existing low-pressure system. The initial pressure and temperature can be so chosen as to provide steam at the desired pressure and temperature to the existing low-pressure turbine.

The curves of Fig. 2 indicate the initial pressure and temperature combinations that will give the desired temperature of steam exhausted from the high-pressure turbine at 200 lb gage pressure. The upper two curves give the initial temperature from the scale at the right

for two desired exhaust steam temperatures of 560 and 600 F. The two lower curves give the amount of power that can be generated with the high-pressure steam at the selected initial high pressure and temperature. Thus, for example, let it be assumed that the existing low-pressure turbine requires steam at 200 lb gage (214 lb abs) and 560 F temperature. The lower of the upper two curves gives the initial pressure and temperature. If the initial pressure is taken as 1200 lb, then a horizontal line drawn from the intersection of the 1200-lb pressure vertical line and the lower of the two upper curves to the scale at the right gives the initial temperature as 930 F. The intersection point of the 1200-lb initial pressure line with the lower of the lower set of two curves and the scale at the left gives the power generated, which in the case under consideration is 45 kw for every 1000 lb of steam per hour. That is, the high-pressure turbine will generate one kilowatt-hour with 22.2 lb of steam.

If the temperature of the exhaust steam from the high-pressure turbine is to be 600 F, then with 1200 lb initial pressure the initial temperature would have to be 970 F, which is too high for the present practice. It would be better to take the initial pressure of 900 lb, which would require only 910 F initial temperature and would be more in line with present-day safe practice. In that case about 39 kw-hr would be generated per 1000 lb of steam per hour.

Generation of Steam and Power in Industrial Plants

In an industrial plant using both steam and electric power the steam can be made at high pressure, used in a high-pressure turbine to generate the required amount of power, and exhausted at the desired pressure in nearly saturated condition into the steam distributing lines. In such cases the power is generated with practically 100 per cent thermal efficiency because there are no condenser losses. The initial pressure and temperature can be selected to balance the power requirements with the demand for steam.

The curves in Fig. 3 can be used to determine the initial pressure and temperature when the steam is exhausted from the turbine at 114 or 214 lb absolute pressure and 5 deg superheat. The required amount of power and steam can be expressed in kilowatt-hours per 1000 lb of steam and applied to the scale at the left of the chart. Assume that the power demand is 3000 kw and the steam demand is 100,000 lb per hr at 214 lb absolute pressure, which is 30 kw per 1000 lb of steam per hour. The lowest curve which shows the kilowatt-hours for exhaust pressure of 214 lb absolute gives the required initial pressure as 900 lb, and the second curve from the top with the scale at the right gives the initial temperature as 670F.

If with the same power and steam demand the exhaust pressure is to be only 114 lb absolute, the initial pressure is found from the second curve from the bottom to be 500 lb, and the initial temperature from the top curve to be 610 F. Thus, by selecting the proper initial pressure and temperature, any desired proportion of power to steam demand can be obtained.

A central steam-heating system can be connected with the power generation business with advantage to both. A certain amount of the heating steam is used a large part of the year and can be considered as a base load. This base load steam can be generated at high initial pressure and at the proper temperature, used in a high-pressure turbine and exhausted to the steam distribution system. Thus, by combining the revenues from power generation and steam distribution the rates can be reduced, thereby stimulating increased uses for both services.

High-Pressure Steam-Generator Design

The general application of high-pressure steam was made commercially possibly by the advance in the art of boiler construction such as welded boiler drums, standardization of high-pressure boiler accessories and pipe fittings, and the development of steel alloys for high-temperature steam generation and usage. There has been also a better proportioning of the various component parts of the steam-generating equipment, so that the modern high-pressure steam generator is a designed unit instead of an assembly of various pieces of equipment made by different manufacturers. The component parts comprise a water-cooled furnace, a boiler, a superheater, an economizer and an air heater.

High-pressure steam has high heat content of the liquid, low heat of vaporization and high heat content of the superheat. Thus steam at 1200 lb pressure and 900 F total temperature contains 1392.7 Btu above the temperature of condensate at 80 F. Of this quantity 523.7 Btu is the heat in the liquid, 611.7 Btu the heat of vaporization and 257.3 Btu that in the superheat. A high-pressure steam-generating unit should be provided with ample facilities for heating water, a comparatively small boiler for evaporating water into steam, and a large

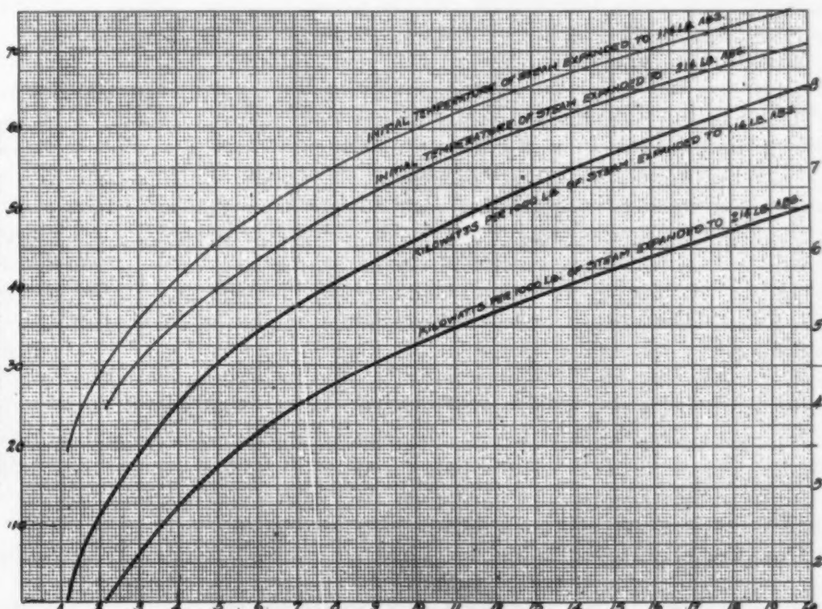


Fig. 3—Initial pressure and temperature combinations with exhaust at 200 and 100 lb gage

superheater. There is usually a large air heater for heating the air employed in combustion.

Heating Feedwater

A large part of the heating of water is done in feedwater heaters with steam extracted or bled from the turbine after partial expansion. The heating of water is accomplished in several steps of nearly equal temperature elevation. The water is heated from 80 to 400 or 450 F. Steam may be extracted from the turbine at 7, 25, 80 and 250 lb absolute pressure. The second stage is usually a deaerating heater. In the heaters the temperature of the water is raised by condensation of the steam. Therefore, stage heating materially reduces the size of the main condenser.

After the last stage of feedwater heating the water flows through a high-pressure economizer which raises the temperature another 75 or 100 deg F. The economizer must of necessity be made of steel tubing capable of standing a pressure about 200 lb in excess of the boiler pressure. It is usually so placed that advantage is taken of the counterflow principle; that is, the hottest water is heated by the hottest gas and the coldest water by the coldest gas. The water usually enters the lowest row of tubes and flows upwards; the gases flow in the opposite direction. The upward flow of water avoids steam pocketing and water hammer.

The Boiler

The boiler is usually extended around the furnace in the form of water-cooled walls. It also serves as a water-cooled frame for supporting the superheater and baffles, and in some designs it forms a water-cooled housing for the economizer. The boiler proper contains only a sufficient number of tubes to protect the superheater against the intense furnace heat, and to supply downcomers for taking the water from the steam drum to the lower drum or headers.

The bent-tube type is generally preferred to the straight-tube boiler because of better water circulation. As the steam pressure rises the specific weight of steam approaches the specific weight of water. Inasmuch as the force producing water circulation is proportional to the difference between the two specific weights, the water circulating force becomes smaller as the pressure rises. The high-pressure boiler must have easy passages for the circulation of water and the flow of steam into the steam drum. Water must be supplied to the evaporating surfaces faster because at high pressures the latent heat is small and more water is evaporated per unit of surface.

The steam drum collects the steam, removes impurities from it and distributes it to the superheater. Many of the bent-tube boilers have two steam drums; one drum collects the steam and separates it from the water. The separated steam then passes to another drum where it is cleaned by washing with feedwater. Clean steam is essential for good operation of a high-pressure turbine. Such drums must be of sufficient size not only for proper arrangement of these steam purifying devices, but also to make the drums accessible for internal inspection and cleaning. These latter features often do not get enough consideration in the general effort to reduce the cost of the steam generating unit. Such features are of particular importance when the feedwater is mostly makeup water, as is the case in many industrial and central heating plants where the exhaust steam from the high-pressure turbine is used for process work or heating purposes and very little or no condensate is returned.

Steam Washing

In high-pressure plants the steam generated must be free from solids to avoid scale deposits in the superheater or on turbine blades. Steam with less than one part per million of solids can be obtained by the use of a steam washer which is a device that brings steam, just before it leaves the boiler, into intimate contact with the incoming feedwater. The steam leaves behind its moisture, consisting of highly concentrated boiler water, and may pick up moisture consisting of feedwater. The concentration of solids in the feedwater is always much lower than that of the boiler water and therefore, any moisture that may be picked up by the steam will contain much less solids than if no washer were used. The water when it enters the washer is at lower temperature and is heated to the saturation point by the condensation of part of the steam. This condensed steam further reduces the concentration of the washer water.

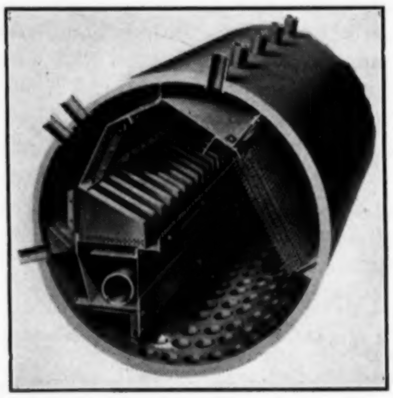


Fig. 4—Bubble type steam washer

Fig. 4 shows a bubble type of steam washer placed in the rear steam drum of a bent-tube boiler. The drum is divided into two compartments by a partition which contains the working part of the steam washer consisting of a row of steam hoods or nozzles the lower ends of which are perforated and immersed in a pan containing feedwater. The steam flows from compartment at the left to compartment at the right through the steam hoods and passes through the perforations in the lower ends of the hoods and through water contained in the spaces between the hoods. Each of these spaces has two opposing jets of steam, one set throwing the water against the other and breaking it into small droplets. These droplets are tossed about by the steam; some of them are thrown up 10 to 12 in. above the water pan and then dropped down to be tossed up again; others are thrown against the sides of the steam hoods and run down over the surface, keeping it continually wet. In other words, there is a vigorous splashing and agitation of the steam and water which brings the two into very intimate contact.

The feedwater is supplied to the trough from which it flows into the washer pan. Part of this water is splashed out of this pan and the rest flows over the weir.

The washed steam passes through the drying screens where the excess moisture is deposited on the wires and is drained off by the upright wires. The washed and dried steam then passes out of the steam drum to the superheater.

Superheaters

The quantity of heat in the superheat of high-pressure and high-temperature steam is large and the heat is at high temperature. The gases which bring the heat to the superheater must be at sufficiently high temperature to contain not only enough heat necessary for the superheat, but the heat must be at high enough temperature to flow into the steam. In other words, the temperature must be such that a good temperature difference is maintained between the gases and the steam during the process of superheating. The greater this temperature difference the smaller can be the superheating surface, and the lower is the cost of the superheater. On the other hand, the hotter the gas when it enters the superheater the more likelihood there is of slag deposition on the superheating surface.

The designer of the superheater has to work between two limits; on the one hand danger of slagging and on the other an excessively large and costly superheater. The slag deposition is a particularly serious handicap with coals having low fusion temperature ash. Lowering the temperature of gases entering the superheater one hundred degrees may reduce the slagging, but it increases the size of the superheater entirely out of proportion to the temperature reduction. It is not only the increased cost but also the difficulty of supporting the heavy superheater and keeping it in alignment that must receive consideration. The superheater is made of small tubes which do not give the elements sufficient rigidity to keep them in alignment. Various spacers and hangers must be devised to keep the elements in their proper position. These spacers and hangers are at higher temperature than the superheater elements and must, therefore, be made of special alloys that will resist oxidation and deformation at these high temperatures. An alloy high in chromium and nickel known as "Fahrite" is generally used for making the spacers and supports for a high-temperature superheater.

In order to increase the effective temperature difference a high-temperature superheater is designed to utilize the counter-flow principle as far as practicable. However, the superheater finishing loops are seldom placed in the hottest part of the gases, because of the danger that the metal might be at times subjected to too high temperature and its life thereby reduced. With a total steam temperature of 900 F the temperature of gases entering the superheater seldom averages less than 1700 F, and may fluctuate over a range of three or four hundred degrees. With a temperature of steam approaching 900 F in the tube and a gas temperature of 1700 F the metal might at times reach temperatures high enough to be damaged. Therefore, the finishing loops are usually placed back of two or more loops containing steam at lower temperature.

It is desirable for a long life of the superheater to keep the temperature of the metal close to the temperature of the steam. This is accomplished by high velocity of steam through the superheater tubes. High velocity requires greater pressure drop which has the additional advantage of more equal distribution of steam through all elements. In a high-pressure and high-temperature installation a reasonably high pressure drop through the superheater is an advantage rather than a detriment.

Metal for the Superheater

The loops of the superheater elements with steam above 850 F are made of alloy steel containing 0.1 to 0.12 per cent carbon, about 0.5 per cent molybdenum, about 5 per cent chromium, and about 0.5 per cent titanium. With steam under 850 F the loops are of carbon steel with about 0.5 per cent molybdenum. The alloys are developed by the tube manufacturers who make available to the superheater designers the physical characteristics at various temperatures. The latter, in turn, select the most suitable alloy for a given temperature service.

To design a superheater for any given steam temperature the designer must know the temperature and the weight of the hot gases entering the superheater, which depend on the rate of combustion in the furnace, the excess air, and the amount of water-cooled surface in the furnace. The temperature also depends on the method of firing and on the adjustment of the burners. A great deal of judgment based on experience with similar designs, checked by actual temperature measurements, is necessary to determine the temperature of gases entering the superheater. The final design is usually the result of number of compromises in the furnace, boiler and superheater designs. The usual tolerance of 10 deg F is very small, considering the many factors affecting the performance of the superheater. It is difficult to appraise properly all these factors in the calculation of performance, in view of which one is inclined to feel that it is a piece of good luck when the actual superheat hits within this small tolerance. The unified efforts of one organization in the coordination of the designs of the component parts of a steam generating unit greatly contributes to the frequency of coming within this tolerance. If furnaces, boilers and superheaters were designed by different organizations, the agreement of the actual with the expected performance would be less frequent.

Superheat Control

The performance of the superheater is important because it is tied to the performance of the turbine. The turbine builder cannot meet his guarantee unless the superheat is right and remains nearly constant over a fairly wide range of load. This nearly constant superheat introduces an additional problem in the design of furnace, boiler and superheater. The wider the range in the steaming rate over which the superheat is to remain nearly constant, the more complicated the design becomes. The temperature of the gases entering the superheater rises with the steaming rate, and consequently the superheat also rises. The following methods are used to keep the superheat nearly constant over a desired steaming rate:

(*a*) A damper-controlled gas bypass which permits a varying volume of the hot gases to bypass the superheater. With this method the superheater is designed for the lowest steaming rate at which the full superheat is desired. As the steaming rate rises above this point, part of the volume of hot gases is made to bypass the

superheater. Thus, the rising temperature of the hot gases is compensated by their decreasing volume. This method of control requires a somewhat larger and more costly superheater, an idle gas pass around the superheater and a damper to regulate the volume of gases flowing through the bypass. The damper and its operating mechanism must be made of heat-resisting metal. The regulation of the bypass damper can be done manually or by an automatic control.

A modification of this method is used in a double boiler with a large superheater on one side and a smaller one on the other side. Varying quantities of gases are passed through the two sides.

(*b*) The superheat is controlled by the manipulation of the fires. In this method the burners are placed at different heights in the furnace. At low rating the burners located in the upper part of the furnace are put in operation and the fires are brought closer to the superheater. At high rates of steaming the lower burners are used. For superheat control over a very wide range of rating this method is used in combination with the bypass damper method.

(*c*) Superheat is maintained nearly constant by the combination of radiant and convection superheaters. The radiant superheater is placed in the furnace as a part of the wall cooling surface, or, part of the superheating surface is placed between widely spaced boiler tubes in the first bank of the boiler. The surface in such locations is exposed to radiation from the furnace and absorbs a greater proportion of heat at low rating. The main part of the superheater is located in the usual place back of the first bank of boiler tubes and absorbs heat by convection. The steam flows first through the radiant section and then through the convection superheater. This method of superheat control is largely a matter of proportioning the amount of surface in the two parts of the superheater. It can be combined with the methods outlined under (a) and (b).

(*d*) Separately-fired superheater. In this method the superheater is set over a separate furnace and the superheat is controlled by the amount of fuel burned. This method has the widest steaming range with constant superheat, but it also is the most expensive and is now seldom used.

(*e*) Control of superheat by partial desuperheating at the higher rates of steaming. In this method the superheater is designed to give full superheat at the desired low rating. At higher ratings the steam is partly desuperheated after it has gone about half way through the superheating process, the extent of desuperheating being controlled by the final temperature. The superheater is installed in two parts with the desuperheating apparatus between them. This method of superheater control is rather complicated and is seldom used in this country.

Air Heaters

The main object of the air heater is to absorb the low temperature heat left in the gases after they have passed through the economizer. The temperature of water entering the economizer in many cases is 400 F or more; therefore, the economizer cannot reduce the temperature of the gases to the point necessary for the high efficiency now required from steam-generating units. The stage heating of feedwater by steam bled from the turbine greatly reduces the effectiveness of the economizer to produce low flue gas temperature. An air heater, on the other hand, with the low temperature of atmospheric air is admirably suited for that purpose. The air heater absorbs the low-temperature heat from the gases and returns it to the furnace with the air used for combustion. In effect, the heat so returned to the furnace raises the temperature of the heat generated by the combustion of the fuel so that a greater percentage of it can be transferred to the steam.

Most of the high-pressure steam generating units have air heaters. The air heater surface costs less than economizer surface, and is more effective because of the low temperature of heat absorbing air. Furthermore, with pulverized coal firing some heated air must be supplied to the mill for drying the coal during the pulverizing process. This coal drying air is a part of the heated air supplied by the air heater.

There are three types of air heaters in use, the plate, the tubular and the regenerative type. With oil firing where corrosion of the air heater is likely to occur the regenerative type is preferred, because there is less corrosion, and the corroded parts are more easily replaced. With natural gas firing, which contains very little or no sulphur, there is usually no corrosion. High sulphur and high hydrogen fuels are likely to cause corrosion of the air heater.

If high efficiency is desired the temperature of gases entering the air heater should not exceed 650 F. The weight of the gases is always greater than that of the air to be heated. The specific heat of the gases is also higher than that of the air. Therefore, the temperature rise of the air is greater than the temperature drop of the gases. Usually the weight of heated air is about 70 per cent of the weight of the gases. The specific heat of the products of combustion, particularly with high hydrogen fuels, is 10 to 15 per cent higher than the specific heat of air. For every 300 deg rise of air temperature there is about 200 deg drop in the temperature of the gases. Thus, if the temperature of the gases is to be reduced from 650 to 350 F, the temperature of the air would rise from 100 to 550 F which is within 100 deg of the temperature of the gases entering the air heater and a very large air heater is required.

With fuels high in hydrogen, the products of combustion contain a considerable percentage of water vapor which is likely to condense on the air heater and cause corrosion. This condensation can be avoided by recirculating part of the heated air, thus raising the initial air temperature above the condensation point. Such air recirculation reduces the temperature difference between the air and the gases and somewhat reduces the effectiveness of the air heater.

Furnace and Burner Design

Most of the high-pressure steam generating units have water-cooled furnaces designed for pulverized coal firing. Such furnaces are also well suited for burning fuel oil and gaseous fuel such as refinery gas, natural gas and blast-furnace gas. Combination burners can be installed to burn any of the three fuels separately or together.

Many of the furnaces for combination fuels employ horizontal firing with the round turbulent burners. Fuel is fed with some primary air through the central part of

the burner, and the secondary air is supplied through a circular set of vanes which give the air a rotative movement. This rotative movement causes the burning mixture to bush out as it enters the furnace and produce turbulence. Most of the mixing of fuel and air occurs near the burners.

There are many high-pressure installations with tangential firing, burning successfully pulverized coal and blast-furnace gas. More recently this method of firing has been extended to fuel oil and natural gas. The fuel is supplied through burners in the four corners in streams tangent to a small circle in the center of the furnace. Most of the mixing is done in the furnace where the streams meet. This method of firing is much less sensitive to the ways in which the air is supplied and smokeless combustion in water-cooled furnaces is very easily obtained.

The furnaces are designed with a combustion space to give rates of heat liberation up to 40,000 Btu per cu ft of combustion space per hour for pulverized coal. The usual rate of heat liberation is 20,000 to 30,000 Btu. The maximum rates of heat liberation are limited by the character of the ash, particularly its tendency to deposit slag on the boiler tubes and superheater. With very fusible ash the lower rates of combustion are generally preferable.

With oil and gas fuels higher rates of heat liberation are practical, in which case the air must be supplied under higher pressure in order that sufficient energy is made available for intensive mixing in the furnace to produce rapid combustion. Intensive mixing is the most important factor in rapid combustion.

Pulverized Coal Preparation

In pulverized coal firing the coal must be pulverized to sufficient fineness to insure nearly complete combustion. It is particularly desirable that the oversize coal remaining on a 50-mesh screen be reduced to a minimum. It is the large particles that pass out of the furnace only partly burned and are deposited in the soot hoppers or flow out of the stack with the gases. Coals with low volatile content must be pulverized to a higher degree of fineness because the fixed carbon burns slowly. Thus a good fineness for high-volatile coal is 65 per cent through 200-mesh, 90 per cent through 100-mesh and 99 per cent through 50-mesh screen. Good pulverization for low volatile coal is 80 per cent through 200-mesh, 95 per cent through 100- and 99.5 per cent through 50-mesh screen. Higher fineness of coal also reduces the amount of slagging that may occur on boiler tubes and superheater.

Fuel Oil Firing

In fuel oil firing the most important factor is good atomization. The oil must be reduced to a fine mist in order that it can be mixed readily with the air needed for its combustion. In oil burners with mechanical atomization the energy needed for reducing the oil to fine mist is supplied as oil pressure at the burner.

The rate of atomization with any given burner tip varies approximately as the square root of the oil pressure at the burner. The minimum pressure at which good atomization of fuel oil can be obtained is about 100 lb. If good operation over a reasonable range of rating is desired, sufficiently high oil pressure must be available at the burner, otherwise the burner tips must be changed too frequently. Oil pressures up to 400 lb are desirable. For good atomization the viscosity of the oil should be reduced to a minimum by heating the oil. With most fuel oils a temperature of 250 to 275 F is sufficient.

Gas Firing

Gaseous fuels are in free molecular subdivision and therefore no pulverization or atomization is needed. The most important factor is to mix the right amount of air with the gas for quick ignition close to the burner and maintain rapid combustion beyond the point of ignition. Many gaseous fuels ignite more readily if only part of the air needed for combustion is supplied in or close to the burner. The remainder of the air is mixed in after the ignition has started. Such progressive mixing can be done readily with tangential firing.

Burning of Acid Sludge

Acid sludge is a refuse from oil purification processes in which sulphuric acid is used to remove tarry or gummy constituents from the oil. Sulphuric acid combines chemically with these tarry compounds which are then easily separated and removed from the oil. The average acid sludge contains about 7.5 per cent of moisture and 7.5 per cent of acid. In some of the sludges these percentages are twice as high. Acid sludge is used as fuel for steaming purposes only in oil refinery plants. It is a waste material hard to depose of. Compared to fuel oil the acid sludge is slow burning and requires higher temperature for ignition. Inasmuch as the products of combustion contain a large percentage of sulphur oxide and water vapor, they are very corrosive. Superheater supports and gas bypass dampers are likely to deteriorate rapidly when acid sludge is burned under a steam boiler. Air heaters corrode rapidly and should not be installed if acid sludge is to be burned. If air heaters are already installed acid sludge should not be burned. Refineries which burn acid sludge recommend low rates of combustion of 15,000 to 20,000 Btu per cu ft of combustion space per hour. They prefer refractory furnaces for sludge burning because they believe that it burns better in such furnaces.

Feedwater

High-pressure plants must have good feedwater, free from scale forming ingredients. Makeup water must be properly treated before it is fed to the boilers. Secondary treatment with phosphate should be used to eliminate the last traces of hardness. Phosphate treatment is usually supplied directly to the boiler to avoid phosphate deposit in feed lines. Phosphate treatment also deposits a film on the heating surfaces which helps protect the metal from corrosion.

Condensate is usually treated with caustic soda to make the water alkaline, or, as it is usually stated to raise its pH value which is usually maintained between 9.5 and 11 in the boiler water. This alkalinity prevents the metal going into solution. Some phosphate is fed to the boiler to eliminate any hardness due to condenser leakage and to form protective film against corrosion. Phosphate treatment is also credited with inhibition of caustic embrittlement. Concentration of

phosphate in the boiler water is usually maintained between 30 and 60 parts per million.

All feedwater must be thoroughly deaerated in a deaerating feedwater heater. In some plants the last traces of oxygen are eliminated chemically by ferrous hydroxide, sodium sulphite or akon. Ferrous hydroxide combines with oxygen to form ferric hydroxide and sodium sulphite combines with oxygen to form sulphate. Akon which is finely divided iron held in suspension combines with the dissolved oxygen in the water to form ferric oxide.

Some operating engineers have their new high-pressure boilers Apexiorized when the installation of the boiler is completed. It is reasoned that the Apexior forms an initial protective film on the inside surfaces and prevents corrosion. As this film is gradually worn off it is replaced by phosphate or oxide protective film. To make Apexiorizing effective the surfaces must be properly prepared; the tubes are pickled in weak sulphuric acid, and welded drums are sandblasted to remove the mill scale. Forged drums are machined on the inside and the Apexior is applied to the machined surface.

Sulphate–Carbonate Ratio Now Employed Less in High-Pressure Plants

There is a strong tendency among high-pressure steam operators to break away from the conventional sulphate to carbonate ratio, particularly when the feedwater is condensate which usually does not contain any sulphate. Some caustic soda is added to maintain the proper pH value. A considerable quantity of sulphates would have to be added to bring the sulphate to carbonate ratio to 3. Such addition of sulphates raises the total concentration in boiler water and may cause trouble from carryover. Many of the high-pressure plants using condensate feedwater keep the total concentration in boiler water below 500 parts per million and some below 200 parts per million.

Recent investigations indicate that sulphates cannot be depended on for inhibiting the so-called caustic embrittlement. Absence of riveted joints with welded or forged drums used in high-pressure boilers, greatly reduces the chances of caustic embrittlement.

In cases where the feedwater consists mostly of treated makeup water, there is usually a high proportion of sulphates and the maintenance of the desired ratio does not involve any hardship. Adding solids to boiler water is always objectionable because they affect adversely the purity of steam. However, sometimes choice must be made of the lesser of two evils. In the treatment of makeup water, chemicals are added to change certain dissolved solids which are very objectionable to other solids less objectionable. For example, sodium phosphate is added to change calcium carbonate to calcium phosphate and sodium carbonate. In high-pressure plants using mostly makeup water, the concentration in boiler water usually exceeds 1500 parts per million.

Creep in Steel

When steel under stresses is maintained at elevated temperature the metal grows in the direction or the action of the stresses. The rate of this growth or creep as it is called, increases as some power of the stress when the temperature remains constant, and also increases as some power of the temperature when the stress remains constant. The rate of creep generally decreases perceptibly during the first 2000 hr but after that initial period becomes nearly constant. The rate of creep in alloy steel is affected by the kind and percentage of the metals of which the alloy is made. Thus molybdenum generally increases the strength of the alloy and thereby decreases the creep. When the stress loading on a specimen is taken off and the temperature is held with zero load, no part of the deformation is recovered, indicating that the creep is permanent.

Most of the creep determinations were made on test pieces in the laboratories where the conditions can be accurately controlled and measurements carefully made.

In the design of high-pressure steam generating units the temperature to which each piece of the equipment is subjected can be fairly closely calculated or estimated. With the temperature known, the rate of creep can be made small by decreasing the stresses. The creep in the boiler parts is very small because the temperature to which the metal is subjected is low, seldom exceeding 650 F. It is appreciable on superheater parts, stop valves and pipe lines because these parts are subjected to higher temperatures. Measurements of the drums of a 1200-lb pressure unit after several years of operation showed no measurable creep.

STEAM ENGINEERING ABROAD

As reported in the foreign technical press

Klip Generating Station

The Annual Report of the Electricity Supply Commission of British South Africa, dated May 8, 1938, contains descriptions of its various power plants and undertakings, notable among which is the reference to the Klip Generating Station, owned by the Commission but operated by the Victoria Falls and Transvaal Power Co. Ltd. This station when completed will have an installed capacity of 424,000 kw and will contain twenty-two boilers each normally rated at 180,000 lb per hr supplying steam at 355 lb gage to twelve 33,000-kw turbine-generators and four 7000-kw house sets. Half of these units are now in service and others will go into operation during the present year. The average load factor for the past year has been 85.1 per cent.

Inasmuch as the station is not located adjacent to an adequate supply of condensing water, cooling towers are provided, as shown in the accompanying aerial view. There will be ten of these towers to serve the completed station, each capable of cooling approximately two million gallons of water per hour from 100 F to 82 F with an average atmospheric temperature of 70 F and 75 per cent humidity.

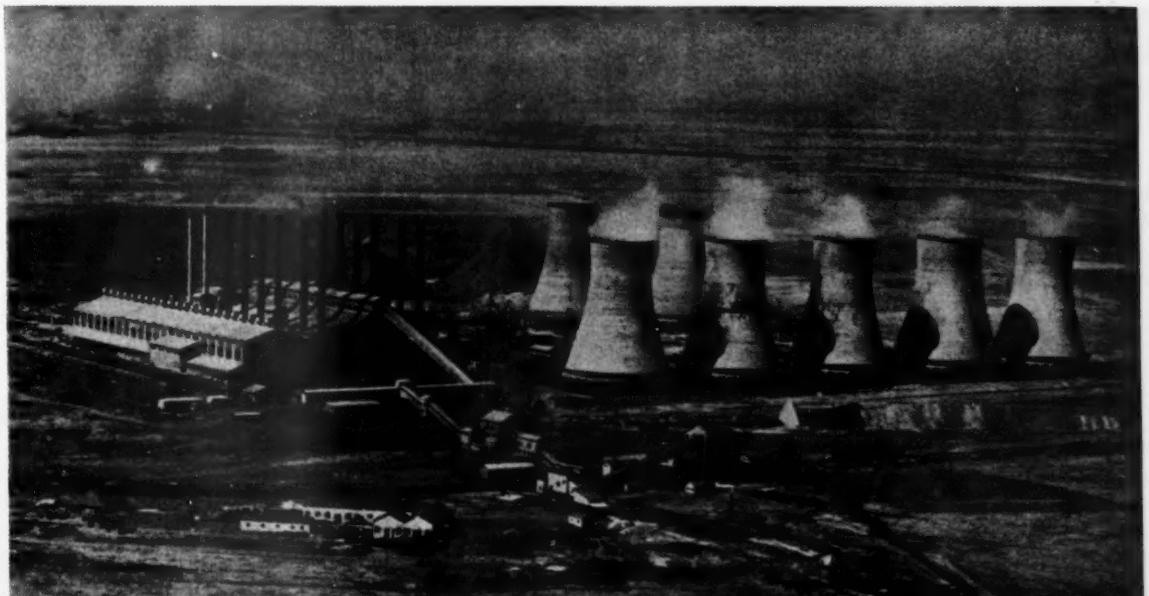
The net capital expenditure on this station to date has been over 21 million dollars.

Notwithstanding the fact that the Klip Power Station will have a capacity of 424,000 kw and is at present only little more than half completed, the growth of load has been so rapid that proposals for the construction of another large power station to meet the anticipated power requirements of the expanding gold mining industry are already under consideration.

New British Station Uses Both Stokers and Pulverizers

The Electrical Times of May 26 contains a preliminary description of the new central station at Littlebrook on the Thames, near Dartford, Kent, construction of which has recently been started.

The initial installation will consist of one 60,000-kw, 1500-rpm turbine-generator and two 30,000-kw, 3000-rpm turbine-generators supplied with steam at 600 lb, 800 F by six 250,000-lb per hr boilers. Three of these will be stoker fired and three pulverized-coal fired. The reasons advanced for employing both types of firing were, greater flexibility in operation, greater economy at varying loads and ability to take advantage of two grades of coal available at the plant. The stoker-fired boilers will have multi-cyclone dust collectors and those burning



Aerial view of Klip Generating Station showing cooling towers and in the foreground the colliery from which coal is supplied directly to the plant

pulverized coal will be equipped with electrostatic precipitators.

In the construction of the building special consideration was given to protection against possible air raids. The turbine room walls are of heavy reinforced concrete construction up to the crane level and all windows are above this. In the boiler room the walls are similar and there are no windows except near the roof. The concrete stacks, $2^1/_2$ times the height of the roof, are exterior to the building. All switchgear and control equipment are to be housed in a separate building, some distance from the plant. The ends of the building have permanent walls and at such time as an extension to the present capacity is contemplated, it will be placed in another building, similar to but removed from the present plant, rather than extending the present building.

Corrosion of High-Pressure Feed Pumps

An article by G. Weyland in the May 21 issue of *Die Wärme* discusses the internal corrosion of boiler feed pumps operating under high pressures, as encountered in some of the newer German installations.

In a number of such cases examination of the pump parts and materials showed them to be free from initial imperfections and similar to those of pumps in other comparable high-pressure plants that had experienced no corrosion troubles. However, comparison of the boiler feedwaters revealed that the pH values in those plants in which pump corrosion occurred was generally around 7 or below, in some cases 6.4 and 6.5, whereas the plants that were free from such corrosion employed a pH of 8 or more.

In discussing the significance of hydrogen ion concentration the author observes that with modern high-pressure steam-generating units the soda content of the boiler water must be kept low to avoid foaming and consequent deposits; likewise blowdown is restricted. In the boiler the velocities are lower than in the feed pump and there is a continuous increase in the soda content and the pH value due to recirculation, which makes the boiler less liable to corrosion than the pump. It is necessary to maintain a definite feedwater alkalinity in order to avoid corrosion. The maximum value at which corrosion will occur in the pump is not pH = 7 but pH = 9.6 at 73 F. Even at higher pH values one may sometimes observe a deposit on the pump surfaces but the corrosive action will be found very minute.

The higher the water velocity through the pump the greater the tendency toward corrosion with a weakly alkaline water. This is particularly noticeable at points where the water undergoes a pressure drop, as at the impeller seals, the bushing sleeves between stages and at the balancing pistons; also, where there is a sudden change in the direction of flow and where eddies occur, particularly at light loads.

The materials heretofore commonly used for high-pressure feed pumps were cast steel, with or without the addition of molybdenum and chromium for the housing, Siemens-Martin, nickel or a special steel for the shaft, cast electric manganese steel for the runner and discharge piece and zinc-free bronzes or monel for the balancing piston and shaft. In the case of low-speed impellers, these and also the discharge pieces may be made of zinc-

free copper alloy. However, for high-speed impellers it is no longer possible to employ bronze, for reasons of strength, and it becomes necessary to use high grade cast-steel alloys. Such materials have proved successful in high-pressure service.

In one case where corrosion had been encountered subsequent trials were made with the runners of four different materials. After 1500 operating hours, part of which was at low loads, the pump was opened and the runners of corrosion resisting materials showed no wear, while those made of cast electric manganese steel, as originally used, were practically destroyed.

The author concludes as follows:

1. The pH must, if possible, be maintained at 8 or over, in which case the usual pump materials may be employed.
2. If, because of certain local conditions, it is necessary to run with a lower pH, down to about 7, then the inner parts of the pump must be constructed of corrosion resisting materials.
3. If the pH is below 7, for even part of the time, the pump housing must also be made of corrosion resisting materials.

New Zealand Station Installs Velox Boilers

An extension to the Evans Bay Station at Wellington, New Zealand, has lately been placed in service, according to *The Electrical Times* of July 7. This comprises a 15,000-kw turbine-generator supplied with steam at 220 lb pressure and 600 F by two Velox boilers, each of 90,000 lb per hr rated output and 10 per cent overload capacity for two hours.

The extension is essentially for peak load and emergency service which was a factor in selecting units of the Velox type in view of their quick starting ability. On a shop test the boilers, containing water at 100 F, were brought up to full steam pressure in $3^1/_2$ min, and on tests made subsequent to their installation, starting with the boilers cold, it was possible to put the turbine on the line in 19 min. This time could have been reduced had it not been necessary to start the boilers consecutively in order not to overload the diesel set supplying the auxiliary power. On test the gross efficiency of the boilers was over 95 per cent and the efficiency curve was very flat down to about one-quarter load. The units are oil fired and very low excess air was maintained.

Feedwater for each unit is normally supplied by a 20,000-gpm motor-driven pump with a Weir steam-turbine-driven pump in reserve. The latter automatically comes into service when the feed line pressure falls below a predetermined value.

The compressor for each boiler, which supplies air under pressure to the furnace, is direct-connected to a gas turbine and through gearing to a 385-hp variable-speed motor. This motor serves the dual purpose of driving the air compressor when starting the unit and thereafter acting as a speed-controlling device whereby the delivery of air is regulated to the requirements of the boiler under fluctuating load conditions. The gas turbine operates at variable speeds up to a maximum of 6000 rpm, and the variable speed of the motor is obtained by means of a Ward-Leonard system supplying variable voltage.

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